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Design, Fabrication, and Testing of a High-Speed, Over-Running Clutch for Rotorcraft

Frank Fitz and Craig Gadd
Epilogics, Inc., Los Gatos, California

August 1998

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Prepared under Contract NAS3–27387

National Aeronautics and
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Design, Fabrication, and Testing of a High-Speed,
Over-Running Clutch for Rotorcraft

by

Frank Fitz, Craig Gadd

Abstract

The objective of this program was to evaluate the feasibility of a very high overrunning speed one-way clutch for rotorcraft applications. The high speed capability would allow placing the one-way clutch function at the turbine output shaft, that is, the input of the rotorcraft's transmission. The low drive torque present at this location would allow design of a relatively light one-way clutch.

During the course of this program, two Mechanical Diode (MD) type overrunning clutches for high speeds were designed. One of the designs was implemented as a set of prototype clutches for high speed overrun testing. A high speed test stand was designed, assembled and qualified for performing overrunning and engagement tests at speeds up to 20,000 rpm. MD overrunning clutches were tested at moderate speed, up to 10,000 rpm and substantial thermal problems associated with oil shear were encountered. The MD design was modified, the modified parts were tested, and by program end, clutches were tested in excess of 20,000 rpm without excessive lubricant temperatures. Some correctable wear was observed and remains as a clutch characteristic which needs further improvement. A load cycle tester with a special, long, sample section was designed, built and then prototype clutches were fatigue tested to verify that the clutch design was suitable for carrying the specified power levels.

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1. INTRODUCTION

Current helicopters use overrunning clutches as critical elements of their speed reducing main transmission systems. Due to clutch mechanism limitations, clutches are located in lower speed but higher torque stages of transmissions. To achieve greater weight reductions in rotorcraft drive system design, it is desirable to locate the overrunning clutch mechanism on the high speed, low torque engine output shaft. Additionally, due to clutch element dynamic interactions, clutches have significant reliability and flight safety concerns

The purpose of this project, an Army funded Small Business Innovative Research (SBIR), Phase II, was to design, prototype and evaluate an aircraft quality lightweight overrunning clutch for operation at speeds from 0 to 30,000 rpm and power levels from 0 to 5,000 hp. The clutch was designed to operate maintenance free for at least 5,000 hours service life and to have high torsional stiffness to help prevent torsional oscillation in the drive train.

An Army funded Phase I SBIR project was completed under Contract NAS3-26921. The objectives of this over-running clutch initial design study program were:

- Define the constraints and characteristics necessary for a successful design.
- Develop one or more design layouts which are likely to meet the requirements.
- Thoroughly analyze the designs utilizing traditional analytical methods.
- Determine and describe in detail a final design based on the analyses.
- Prepare a Final Report which summarizes the results of the program and includes a detailed Phase II proposal.

The Phase I program was completed successfully and an overrunning clutch design was produced, including preliminary part drawings. The following list describes the minimum operating conditions for this design.

Speed Range:	0 to 30,000 rpm
Maximum Continuous Power:	5,000 horsepower
Maximum Continuous Torque:	10,500 in-lb
Limit Torque (200% max):	21,000 in-lb

Yield Torque (230% max):	24,100 in-lb
Ultimate Torque (300% max):	31,500 in-lb
Temperature Range:	-65°F to 400°F
Lubricant:	MIL-L-23699
Maintenance Free Service Life:	5,000 hours
Oil-off Life (@ 70% power)	30 minutes

Maximum Continuous Torque is the torque level which the clutch can operate at for the full service life. Limit Torque is the torque level which the clutch can reliably operate at for short and infrequent periods of time,. Yield Torque is the torque level which the clutch can achieve without exceeding the yield point of any materials. Ultimate torque is the torque level which the clutch can achieve without fracturing any materials.

A Mechanical Diode (MD) style one-way clutch was designed for this application. This type of one-way clutch is in production for automotive automatic transmission applications; however, special bearing and lubrication provisions were made for high speed operation. A cross section of the design is shown in Figure 1.

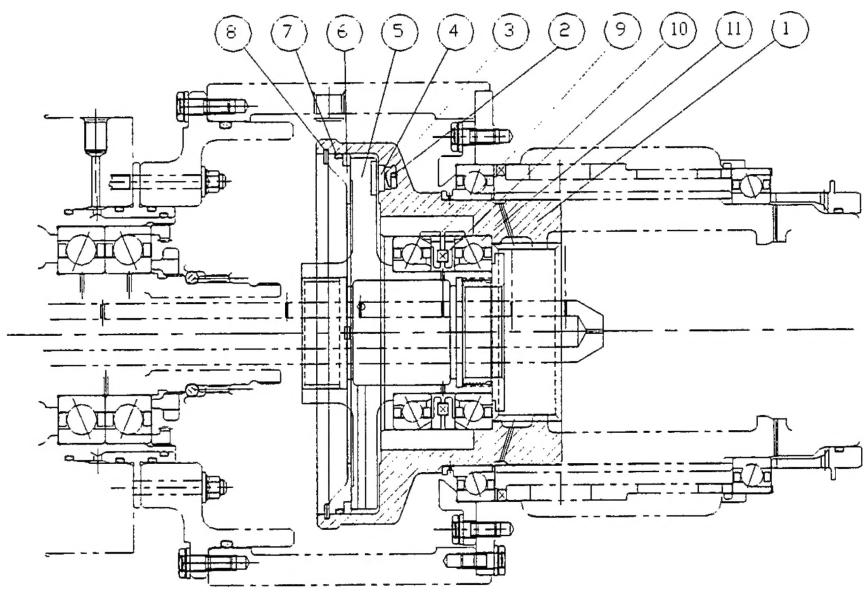


Figure 1

Item 1, P/N 100869, Pocket Plate - Provided an 80 mm bore to fit Bell Helicopter Textron Inc. (BHTI) furnished bearings. This part is the main structural part of the one-way clutch and houses struts (the locking element of the clutch), springs and counterweights to assist high speed engagement.

Item 2, P/N 100546, Torsion Spring - This provides approximately 0.013 lbf bias on the

strut toward the notch plate, item 5, at all speeds. Our standard spring for normal MD designs is used. It has proven adequate at speeds up to 7,000 rpm.

Item 3, P/N 100871, Cam – Assists strut movement into an engaging position at high speeds. Low speed (10,000 rpm or lower) MD designs use only springs for urging the struts into locking position. However, since this design required reliable lockup at clutch speeds approaching 30,000 rpm, an angular velocity actuated cam was added to this design.

Item 4, P/N 200213, Strut – A "T" shaped steel plate which engages notches in the notch plate, Item 5, during lock up and folds down into the pocket during overrunning. The "T" projections have been lengthened and made symmetrical compared with conventional (automotive) designs. The greater length of these projections allow portions of them to lie adjacent to continuous portions of the Notch Plate, Item 5, surface. This prevents the "T" end of the strut from entering an engagement notch during overrun.

Item 5, P/N 100865, Notch Plate – This is the second major component of the clutch and contains a series of notches formed in one face. These notches allow one-way lockup to the pocket plate by means of the struts, Item 4. Three 1/8 inch diameter radial holes through the flange provide positive oil flow to the region of the struts and notches at the outer periphery of the part.

Item 6, P/N 100872, Backing Plate – Serves to support axial spreading forces during lockup and retains oil in the strut region during overrunning.

Item 7, O-ring – A standard, commercially available Viton ring.

Item 8, P/N WH-590, Retaining Ring – A commercially available retaining ring with balancing slots.

Item 9, P/N 200216, Spacer – Provides bearing preload management.

Item 10, P/N 1249-15, Wave Spring - This part has been added on recommendation of BHTI to properly control the Bell Helicopter Textron, Inc. (BHTI) provided bearings.

Item 11, P/N 599-322-316-101, Bearing - This part is the BHTI recommended and provided bearing.

Item 12, P/N 200214, Retainer - This part is a threaded retainer to locate the bearings, Item 11.

Weighted cams were included in this design to assist strut actuation. The strut used in this application would push against its pivot pad with a force, Fd, of 306 pounds at 30,000 RPM. The frictional drag resisting strut pivoting with an average pivot pad radius of 0.067 inch and assuming a co-efficient of friction of 0.1 would be 2.28 in-lb which is given the label Tdrag.

Where:

f	=	coefficient of friction	
ω	=	rotational speed in radians/second	
strutmass	=	mass of strut - mass of displaced oil	
Fd	=	strutmass $\cdot$ Rs $\cdot$ ω^2	Equation 1
Tdrag	=	Fd $\cdot$ f	Equation 2

Given the small strut lever arm, Rl, (0.20 inch), the spring force required to generate a counter force to Tdrag would be 11.4 lbs. This much spring force causes concern over low speed overrunning wear and high drag during high speed overrun.

This cam finger arrangement is designed to augment the normal strut actuation spring with a force proportional to the rotational speed of the pocket plate side of the clutch. In this way the problems with low speed wear and high speed drag are avoided. At low speed, little added force is generated. At high speed overrun, the pocket plate, which is connected to the engine, is not rotating and therefore not causing the cam fingers to generate additional force. Tdrag and the additional force generated by the cam finger are both functions of rotational speed and closely parallel each other over the speed range.

The cam finger is supported outboard by a pivot pocket formed in the pocket plate which is the driven member of the clutch. This cam finger is located under the strut and is shaped so that its mass center is offset from its pivot in the direction of the strut. As the strut and cam assembly rotate at engine speed, the offset of the cam finger generates a torque proportional to the rotation speed. This torque on the finger causes the cam surface of the finger to push on the bottom of the strut, overcoming Tdrag and biasing the strut towards engagement with the notch plate.

Tstrut	=	(Fs $\cdot$ Rl) - Tdrag	Equation 3
Fp^2	=	$Fs^2 + ((Tf/Rc \cdot \sin\beta) + Ff)^2$	Equation 4

$$Tf = \Sigma Tn \qquad \text{Equation 5}$$

$$Fs = Tf/Rc \cdot \cos\beta \qquad \text{Equation 6}$$

Tf is a summation of torques calculated from a discrete element model of the irregular profiled cam finger.

$$Fn = Rn \cdot Mn \cdot \omega^2 \qquad \text{Equation 7}$$

Where Mn is the mass of element "n"

$$Tn = Fn \cdot Ln \qquad \text{Equation 8}$$

Another design consideration for this application was lubrication and heat generation during overrunning. A simplified model of the MD was studied to estimate internal temperature during overrunning conditions. The model used was two annular plates, one spinning, one stationary. Plate ID was 2.5 inches and plate OD was 5 inches. Lubricant was considered to be supplied to the gap between the plates at the ID and to leave the gap freely at the OD of the plates. Specific heat of the lubricant was taken to be 0.5 BTU/lb-F and the thermal conductivity of the steel plates was taken to be 26 BTU/hr-ft^2-F. The forced convection heat transfer coefficient at the oil-plate interface at the velocities involved was considered large compared with the thermal conductivity of steel. Thermal properties were considered constant over the temperature range of interest. Actual variation over the practical range of interest was estimated to be less than 10%. Inlet oil temperature used for this analysis was 120F.

An energy balance on this model yielded the following.

Energy in at the lubricant inlet, the specific heat times temperature times mass flow:

$$q_{in} = C_p T_{in} dm/dt \qquad \text{Equation 9}$$

Similarly, energy out at the lubricant outlet is specific heat times temperature times mass flow:

$$q_{out} = C_p T_{out} dm/dt \qquad \text{Equation 10}$$

Energy conducted out by the plates is twice the surface area of the plates times the conductivity of steel times the temperature difference across the plates divided by plate

thickness:

$$q_{cond} = 2\,A\,K_s\,(T_{avg} - T_w)/t_p \qquad \text{Equation 11}$$

Where

$T_{avg} = (T_{in} + T_{out})/2$ and T_w is the outer wall temperature of the plates

Energy generated in the gap between the plates was computed from the previous work which predicted overrunning torque and horsepower. The conversion is 2546 times horsepower equals energy in BTU/hr.

$$q_i = 2546\ HP \qquad \text{Equation 12}$$

Lubricant flow through the MD which was previously calculated was used for the thermal analysis except that provision was made to also include forced flow in addition to that naturally developed by the spinning plate.

An energy balance on the model provides thermal estimates:

$$q_{in} + q_i - q_{cond} - q_{out} = 0 \qquad \text{Equation 13}$$

Energies and temperatures were computed for speeds from zero to 30,000 rpm, using only the "self pumped" oil flow which is essentially linear with speed, with a predicted flow of 0.22 gpm at 30,000 rpm. This resulted in extremely high predicted lubricant temperatures.

Increasing the flow through the MD model 9X to 2 gpm provided more useful results. The lubricant outlet temperature increased exponentially as speed increased to an ultimate value of 300F at 30,000 rpm. Overrunning tests carried out on much smaller, automotive torque converter compare favorably with predictions.

The major objectives were to demonstrate the high speed capability and high reliability of the MD and to provide a final design which can be integrated in the design of rotorcraft drivelines. To accomplish this, prototypes were built and tested, based on the concept designs developed in the Phase I program. The major Phase II program objectives were the following:

- Complete a detailed mechanical design of a high speed over-running clutch based on the design concept and analysis developed in the Phase I program.
- Fabricate prototypes to demonstrate the mechanics and geometry of the design.

- Conduct high speed tests from 0 to 30,000 rpm to demonstrate high speed capability.
- Conduct life cycle tests to demonstrate high reliability.

2. DEVELOP AND EVALUATE THE MECHANICAL DESIGN

A preliminary layout of the MD in the Bell Helicopter Textron, Inc. (BHTI) test rig was developed and then revised. The revised version included BHTI supplied parts as well as the MD installation. The central portion showing the MD to fixture relationship is shown in Figure 2.

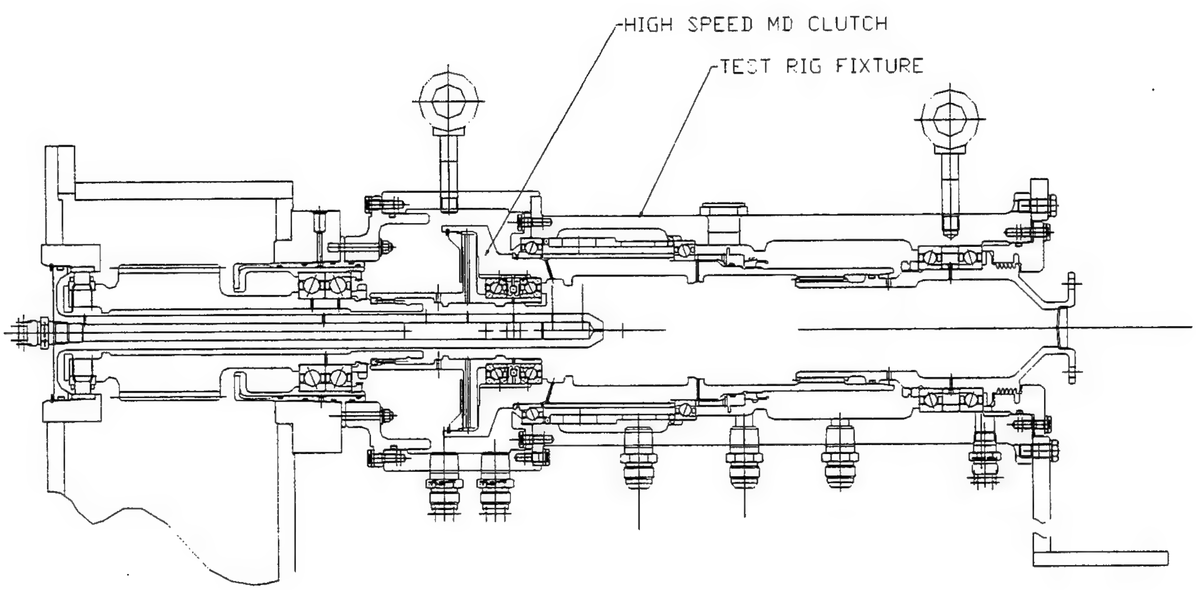


Figure 2

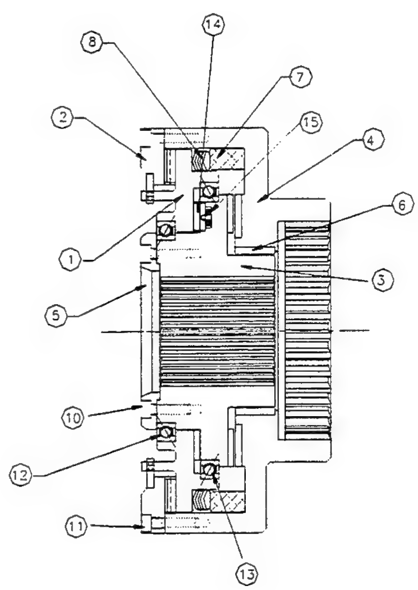
Compared with the original design, the two main rotating parts were lengthened and portions resized to make spline connections with driving elements of the test

fixture; the outside diameter of the part which house the locking struts was modified to fit support bearings in the fixture, and the bearing arrangement between the two parts was modified to use high speed bearings furnished by BHTI along with changes to improve the bearing support.

These changes suggested by BHTI were adopted essentially as layed out and the prototype design was developed to fit this BHTI layout exactly except that the two primary rotating parts have been shortened to allow installation in an in-house, existing overrun test fixture.

Design calculations and estimates performed in Phase I were reviewed; and in those cases where dimensional changes occurred, calculations were revised. Original estimates of the validity of this design appeared to remain valid.

An alternate design concept for a high speed MD based one way clutch was conceived early in the Phase II effort. This design was carried forward through the prototype stage of the project to serve as a back up concept to minimize program risk. This design is a cam actuated, self synchronizing dog clutch which we are designating a Cam-MD. A description of the mechanism follows.



Parts List:

1. Notchplate/Follower
2. Cam Plate
3. Pocketplate/Follower
4. Dog Cover
5. Thrust Plate
6. Bushing
7. Spring Plate
8. Wave Spring
9. Extension Spring
10. Screw
11. Screw
12. Bearing
13. Bearing
14. Strut
15. Spring

Figure 3

The mechanism shown in Figure 3 consists of the following components: Fixed by bolts (11) to the Dog Cover (4) and coaxial with its centerline, is the Cam Plate (2). The Cam Plate contains five discrete cam surfaces which are machined as recesses into the surface of the plate. The cam features are equispaced and on a diameter

which is radially inward from the bolts (11) securing the assembly.

Matched to the individual cam surfaces are individual cam followers. These features extend axially from the Notchplate/Follower (1) and make contact with the individual cam surfaces. The Notchplate/Follower is coaxial and concentric with the cam plate. Coplanar but offset and opposite the cam followers are Mechanical Diode style notch features. These features are equispaced and on a diameter which is radially inward from bearing (13).

On this same diameter but opposed from the notch features are spring actuated struts (14) and springs (15). Both the struts and springs are carried in machined reliefs which Epilogics typically refers to as "pockets". These pockets are machined recesses in Pocketplate/Follower (3). Coplanar but offset and opposite the pocket features are outwardly facing dog teeth which face the dogs from the Dog Cover.

The mechanism is designed to allow for relative rotation between the input shaft (3) and the output shaft (4) as shown in Figure 4, in only one direction. Relative rotation can occur when the Pocketplate/Follower is rotating in a direction so that there is no load acting on the face of the strut. This allows rotation of the Pocketplate/Follower to occur relative to the inner Notchplate/Follower, the Dog Cover and the Cam Plate.

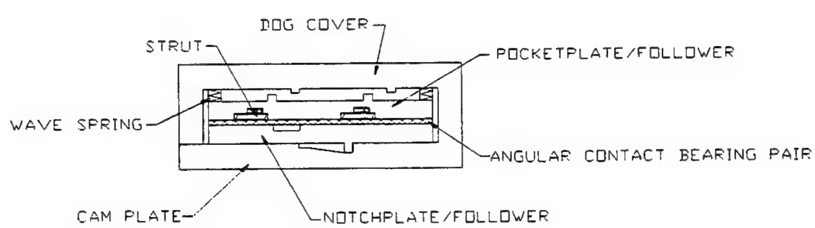


Figure 4

When relative rotation between the Pocketplate/Follower and the Notchplate/Follower is zero, torque can be transmitted between from the input shaft to the output shaft. For this to occur, the following events must take place.

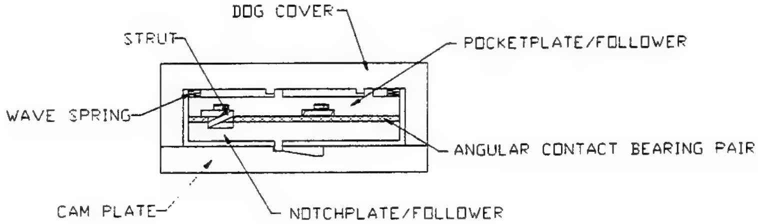


Figure 5

The strut in Figure 5 must be must be loaded so that relative angular velocity can occur between the Notchplate/Follower and the Cam Plate. As rotation between these two components occurs, so does translation of the NotchPlate/Follower and Pocketplate/Follower. The means of this translation is the camming action of the Cam Plate acting on the Pocketplate/Follower. As translation occurs so does compression of the Wave Spring shown in Figure 5, situated between the Pocketplate/Follower and the Dog Cover. As the wave spring is depressed dog teeth projecting from the Pocketplate/Follower are driven precisely into engagement with the dogs projecting from the Dog Cover. When contact is made between the dog features, torque can be transmitted between the input and output shafts.

The resolution, angle between engagements, of the prototype clutch with three pockets and five notches is:

$$Resolution = \frac{360^o}{(5Notches)\,x(3Pockets)} \qquad \text{Equation 14}$$

$$Resolution = 24^o \qquad \text{Equation 15}$$

The force that the strut exerts due to centrifugal acceleration at 30,000 RPM and a radius Rs = 2.00 in. is given below. Using a lightweight titanium strut and neglecting oil buoyancy effects, the centrifugal force is as follows.

Variables:

Ms = Mass of the strut = .00143 lbm
Rs = Radius to the strut = 2.00 in

ω = omega = 3135 rad/sec^2
g = Acceleration of gravity = 386 in/sec^2

$$Centrifugal\ Force = \frac{Ms \bullet \omega^2 \bullet Rs}{g} = 72lbf$$ Equation 16

The cam follower position equation was derived using polynomial methods as outlined in Design of Machinery, McGraw-Hill, Copyright 1992. A brief outline of the design analysis is described below.

In order to ensure reasonable behavior of the cam follower, the following six design constraints were imposed upon the position velocity and acceleration equations shown below.

1. $\beta = 0$, $y = 0$ 4. $\theta = \beta$, $y = .1$

$$y = C0 + C1 \bullet [\frac{\theta}{\beta}] + C2 \bullet [\frac{\theta}{\beta}]^2 + C3 \bullet [\frac{\theta}{\beta}]^3 + C4 \bullet [\frac{\theta}{\beta}]^4 + C5 \bullet [\frac{\theta}{\beta}]^5$$

Equation 17

2. $\beta = 0$, $v = 0$ 4. $\theta = \beta$, $v = 0$

$$\upsilon = \frac{1}{\beta}[C1 + 2C2 \bullet \frac{\theta}{\beta} + 3C3 \bullet \left[\frac{\theta}{\beta}\right]^2 + 4C4 \bullet \left[\frac{\theta}{\beta}\right]^3 + 5C5 \bullet \left[\frac{\theta}{\beta}\right]^4]$$

Equation 18a

3. $\beta = 0$, $a = 0$ 4. $\theta = \beta$, $a = 0$

$$a = \frac{1}{\beta^2}[2C2 + 6C3 \bullet [\frac{\theta}{\beta}] + 12C4 \bullet [\frac{\theta}{\beta}]^2 + 20C5 \bullet [\frac{\theta}{\beta}]^3]$$

Equation 18b

Solutions to the constant coefficients were then obtained:

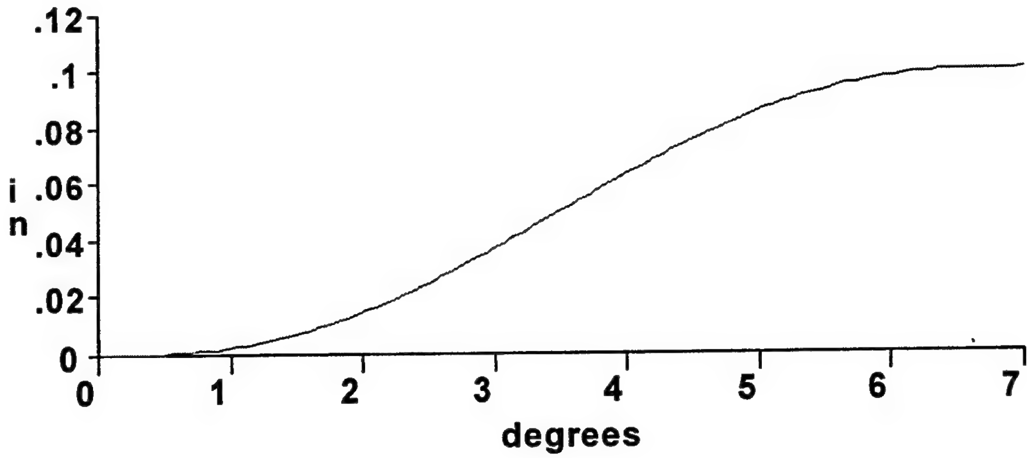


Figure 6 - Translation (y) vs Angular Displacement (theta)

C0 = 0
C1 = 0
C2 = 0
C3 = 1
C4 = -1.5
C5 = .6

Variables:

θ = angular position. (degrees)
β = final angular position. (degrees)
y = follower translation. (in.)
v = velocity of follower translation. (in/degree)
a= acceleration of follower translation. (in/degree^2)
C0, C1, C2, C3, C4, C5, C6 =constants (unitless)

To verify that boundary conditions had been satisfied , plots were made for the position and acceleration curves. Figures 6 and 7 display the results.

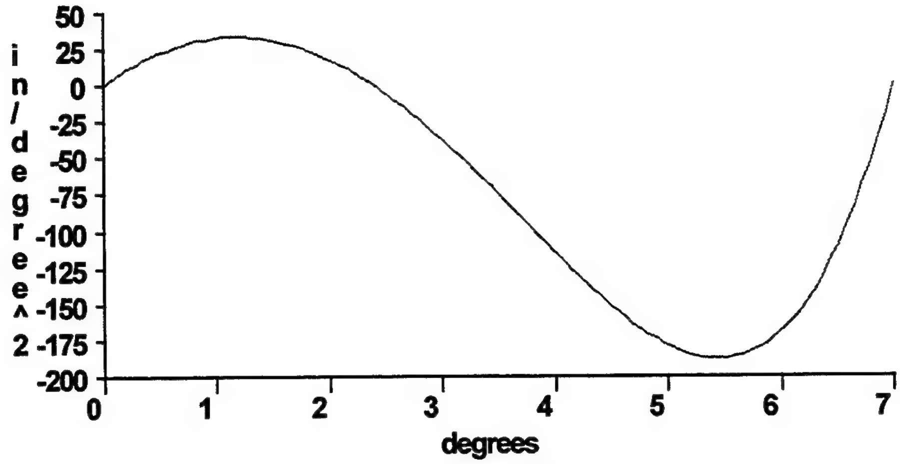


Figure 7 – Axial Acceleration (a) vs Angular Displacement (theta)

Cam Pressure Angle - In order to ensure that the cam follower would easily translate, a cam pressure angle of less than 30 degrees is considered acceptable. The cam pressure angle is defined as the angle between the axis of follower translation and the normal to the cam surface and is given by the relation.

$$\phi = \mathrm{atan}((v/N)/(y/N + r)) \qquad \text{Equation 19}$$

where,

v = angular velocity of the cam (in/sec)
N = translation velocity of the cam (rad/sec)
y = follower displacement
r = cam base circle radius
trial and error methods were used until suitable values for r and b were found. The results are as follows.

$r > 2.4$
β>6.5 degrees.

Figure 8 depicts the results.

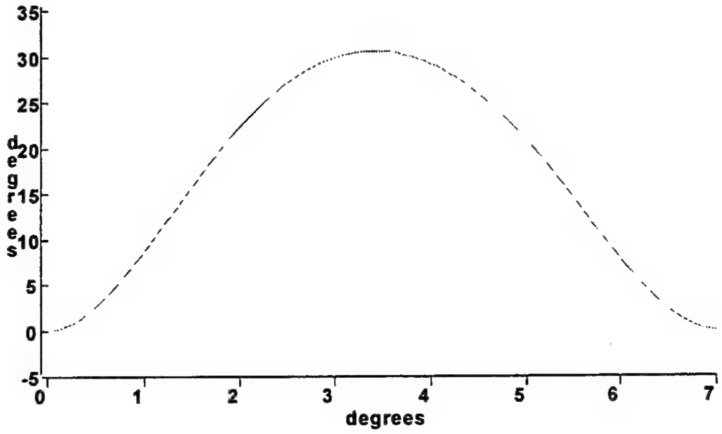


Figure 8 – Pressure Angle phi vs theta

Dynamic Force Analysis

The dynamic force analysis was based on the model shown in Figure 9.

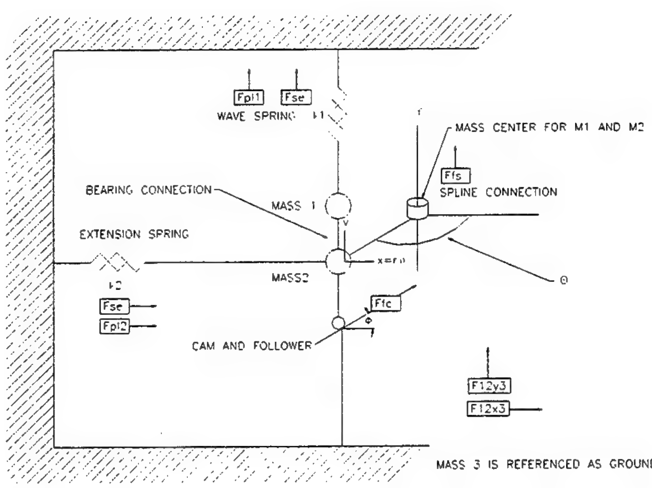


Figure 9 – Spring-Mass Diagram

The point masses m1 and m2 represent the masses of the Notchplate/Follower and the Pocketplate/Follower respectively. The mass m3 represents the Dog Cover and the Cam Plate, which for the purpose of simplifying the model has been shown as

ground. The springs, k1 and k2 represent the spring constants for the wave spring and the coil spring.

To allow for angular acceleration to be applied to the cam as well as variable angular velocity, the polynomial equations and derived coefficients needed to be rewritten in differential notation, the form of the equation is shown below.

$$y = f(\theta) \qquad v = \frac{dy}{d\theta} \bullet \frac{d\theta}{dt} \qquad a = \frac{dy}{d\theta} \bullet \frac{d^2\theta}{dt^2} + \frac{dv}{d\theta} \bullet \left(\frac{d\theta}{dt}\right)^2$$

Equations 20 & 21

Equations were then written to describe internal forces within the clutch. Where the variables a, v, y now represent the differentials above.

F12x3 = Ffct*cos(ϕ)+(k2*r*θ)+Fpl2
F12y3 = Ffct*sin(ϕ)+Ffs+m12*a+(k1*y)+Fpl1
Ffct = u1*(F12y3*cos(ϕ)+F12x3*sin(ϕ))
Ffs = u1*(F12x3)
ϕ= atan((v/N)/(y/N + R)
N = SQRT(2*d2theta*(theta))

Variables:
F12x3 = force of mass 1 and mass 2 on mass 3 in the x direction. (lbf)
F12y3 = force of mass 1 and mass 2 on mass 3 in the y direction. (lbf)
Ffct = frictional forces on the cam follower interface. (lbf)
Ffs = frictional forces acting on the spline during translation. (lbf)
Φ = pressure angle (degrees)
N = rotation velocity of mass 1 and 2 relative to mass 3. (radians/sec)

Constants and Knowns:
d2theta = angular acceleration of mass 1 and 2 relative to mass 3. (radians/sec^2)
u1 = coefficient of friction between oiled steel on steel
Fpl2 = Force of the spring2 preload. (lbf)
Fpl1 = Force of the spring1 preload. (lbf)
m1,2 = mass of the follower (lb-sec^2/in)
a = translation acceleration of mass follower (in/sec^2)

y = translation of the mass 1,2 follower (in)
v = velocity of the mass 1,2 follower (in/sec)
R = radius to the cam follower interface
theta = angular displacement of cam shaft. (radians)

With the dynamic equations programmed using commercially available math software [1], trial and error methods were used to design necessary springs, determine axial and radial bearing forces as well as find limitations in part geometry and strength.

Considered to be of prime importance in the sizing of components was the contact stress between the follower and the cam surface. To determine contact stress values for radius of curvature were determined using the equation shown below.

rhop = ((r + y)^2 + (v/N)^2)^1.5/((r+y)^2 + 2*y^2 - ((a/N^2)*(r+ y)))

Equation 22

Variables:
rhop = radius of curvature (in)
(remaining variables and units are same as those in above list)

Values for radius of curvature were then plotted for the range of theta, to find minimum and maximum values of radius which would yield the greatest contact stress. The plot depicting radius of curvature is shown as Figure 10.

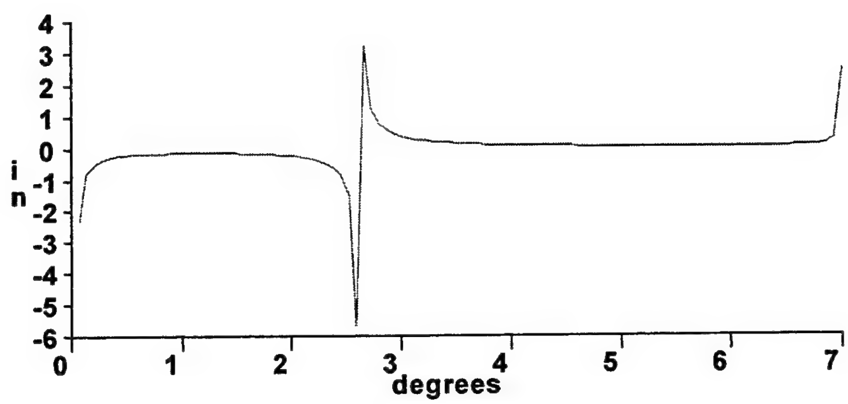


Figure 10 – Radius of Curvature vs Angular Displacement (theta)

Using Roark's Formulas for Stress & Strain [2] (Table 33. Loading case 1b, 1c) maximum contact stress for a sliding follower with a .040 in radius and a .75 in length was determined to be

$$Sc = 50{,}000 \text{ psi.}$$

Two torque capacities were calculated for the clutch. The first capacity being the stress that the dogs would be able to withstand during engagement, and the second capacity being the stress that the strut would be subject to during approximately 7 degrees of preengagement.

Neglecting the shock to the dogs, and assuming a shear loading condition at the base formulas for torque and shear stress are as follows.

$$\tau = F*r \qquad \sigma = F/A$$

Equations 23 & 24

τ = total torque applied = 1,000 ft*lbf
r = radius to the center of the dogs = 1.891 in
F = force collectively applied to 15 dogs = 6345 lbf
A = total area in shear for 15 of dogs = 1.00 in
σ = 6345 psi

Compressive stress induced on the strut was found using the previously derived dynamic equations. The figure shown below is for a 3,000 RPM/S engagement with the same strut dimensions referenced in the centrifugal force calculation.

Because the dog teeth should not produce a resultant tangential force, load on the plain bearing should be zero during engagement and minimal during the pre-engagement.

Both designs were evaluated carefully and ultimately the counter weight type design was selected for the following reasons:

Simplicity – The design has many fewer parts and is less dependent on careful forming of features and on close tolerances.

Risk – Both designs offer considerable risk since neither had ever been evaluated at extreme overrunning speeds. However, since designs similar to the counter weight version had successfully functioned at

conventional overrunning speeds, this design was felt to offer slightly lower risk.

3. Sample Preparation

Primary design drawing packages were distributed to several candidate machining vendors as part of a solicitation for quotations and comments. As a result, several minor changes were made to the drawing package to enhance completeness and manufacturability. A machining vendor was chosen and an order was placed.

Prototype parts were produced as specified in the drawing packages with the exception of the primary design Pocket Plate (P/N 100869) and Backing Ring (P/N 100872). These two parts were modified to allow using a Woodruff Key based anti-rotation provision instead of the originally specified machined lug approach

All parts received appeared to conform to basic specifications of the drawing packages and were successful assembled without further modification.

Prior overrunning tests had been run in-house on drag racing torque converter MDs and on various automotive and industrial prototype one way clutches. The helicopter prototypes were both longer and somewhat larger in diameter than those which were normally tested. Therefore, the test fixture required modification.

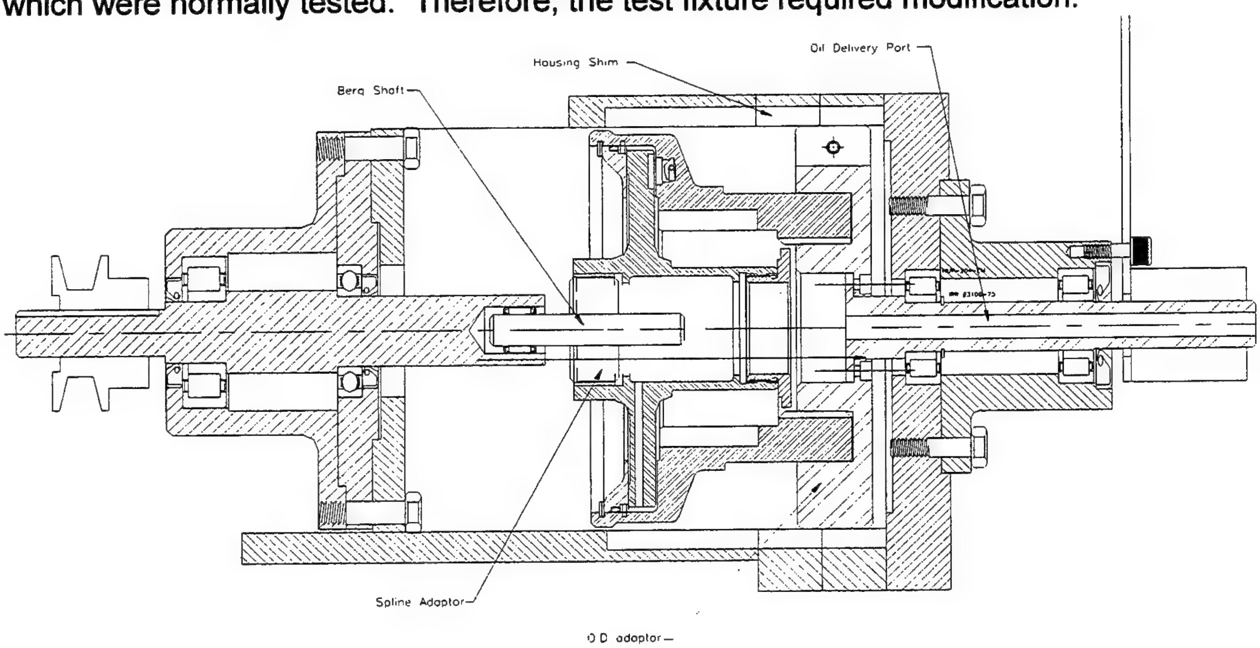


Figure 11 – Overrun Fixture Cross Section

A cross sectional view of the fixture is show in Figure 11. A pair of extenders for the fixture, labeled Housing Shim were added to provide axial space and the oil delivery system was modified from the existing ported bulkhead type of feed to an centerline stationary tube type of feed. Initial operation of the fixture revealed serious oil leaks at the shim joints and a need for a closed end with cross holes modification to the centerline oil feed tube. These deficiencies were remedied and the fixture was prepared for overrun testing.

The primary design prototype was assembled, mounted in the overrunning test fixture and operated at speeds up to 3,500 rpm. Testing was brief due to several oil leaks caused by inadequate sealing of the newly modified test fixture. The prototype was then disassembled and inspected. Visual appearance of the MD internal parts was unchanged, suggesting that low speed operation was as designed.

A full dimensional check of the primary prototype was performed. Parts included in this inspection were:

100546	Spring
100865	Notch Plate
100869	Pocket Plate
100871	Cam
100872	Backing Ring
200213	Strut
200214	Retainer
200216	Spacer

Critical dimensions were found to be generally within specification and based on this inspection, the parts are believed suitable for low speed, general function testing. Sufficient discrepancies were found that enhanced in-process controls were used for subsequent prototype fabrication.

The modified overrunning test fixture was reworked to eliminate oil leaks. During repairs, the oil delivery tube which is located on the tester's central axis was modified to increase the size of the oil holes.

The primary prototype was disassembled, the bearing retaining nut was securely tightened and secured with thread locking compound. On final assembly, the O-Ring seal at the OD of the oil retaining washer was coated with RTV silicone rubber compound to insure a good oil seal. No part modifications were required for proper assembly.

On completion of preparations, a 48 hour overrunning test was conducted. Initial

operating speeds were between 4,800 and 5,000 rpm with the speed ultimately increased to 7,200 rpm. Higher speeds were not possible due to the horsepower limitations of the engine lathe which is the power source for the overrunning tester. The 48 hours of running time were accumulated over a period of 6 days. Lubricant temperature varied with ambient temperature but was generally about 200°F. No unusual behavior was observed during the test.

The primary prototype was completely disassembled on completion of the overrunning test and each component was inspected for signs of wear or distress. Results were as expected for an MD type one way clutch; no signs of wear or distress were discovered. The struts and notches were examined especially carefully. Some overrun related abrasions were evident on both notches and struts; however, this slight level of abrasion was caused by hand turning the MD during assembly and preparation since the appearance of these parts after the overrunning test was the same as before the test. This is typical of MD's which are overrun with liquid, in this case Automatic Transmission Fluid (ATF), present in abundance at the strut/notch plate interface since the struts are maintained out of contact with the notch plate by fluid forces.

Nominal lock up behavior was verified by manually loading the reassembled MD in the lock direction with a torque wrench. Wrench capacity is 250 ft-lb and it was operated in a 50 ft-lb overload condition to apply a 300 ft-lb torque to the MD. Approximately 2,000 loadings were applied over a 3 hour period. On completion of loading, overrunning remained smooth and quiet. Subsequent inspection revealed no problems.

Minor changes were required to conform to the BHTI high speed tester design. Revised drawings were forwarded to BHTI for review and correction if necessary to insure proper interface with the high speed tester. A set of the final drawings as approved by BHTI and provided to fabricators is attached. (See Appendix B)

This program was originally developed assuming that the prototype material would be S7 tool steel or a similar material. However, some of the prototypes were fabricated out of a precipitation hardening steel from Carpenter, designated AerMet 100. This material presented at least two challenges. First, the material cost was significant, over $19 per pound for the AerMet compared with under $4 per pound for S7. AerMet was available in 8" diameter and a length of approximately 20" was required for the two major parts. This resulted in an order of magnitude cost increase. Machining costs were estimated to be higher for the AerMet also, due to its increased hardness in annealed condition compared with S7. A small test bar of AerMet was used for rough material evaluation. Initial tests indicated that all of the small end mill work presently required on the notch and pocket plates would need to be done by plunge EDM (Electrical Discharge Machining). Further machining tests were done on the test bar to better determine what portion of the metal removal could be done by

milling and what portion by EDM.

A local vendor experienced in heat treating AerMet 100 was located and the heat treating procedure for the prototypes was determined. Although the AerMet is more stable through heat treat than S7, we decided to leave the fabrication process, including grind stock allowances the same for both materials. This increased the metal removal required during final grinding operations slightly, but was a safe approach.

Machining of these complex parts in general and the AerMet 100 parts specifically proved to be a challenging task for local machining services vendors. Ultimately the task was completed and the parts were produced within specification and with excellent surface finishes and overall appearance.

4. Demonstrate High Speed Capability

A comprehensive test plan was developed to evaluate the MD's overrunning performance at high speeds. A copy of the plan is attached as Appendix A. The sequence of testing was to perform the planned tests at incremental speeds up to 20,000 rpm. Then, if all was going well, to perform the 30,000 rpm tests on a best efforts basis.

Twenty seven high speed tests were performed and results of these test runs are summarized in Table1 below. Following the table is a discussion of the significant groups of tests and the related development of the MD.

Table 1 – High Speed Overrunning Test Summary

				Speeds: (rpm)			Flow (gpm)	Temp (°F)		
Run Nr	Date	Events	Duration Minutes	Notch Plate (I/R)	Pocket Plate (O/R)	Differ-ence	Oil In	Oil In	Oil Out	Delta
1	5/22/96	Warmup,		10000	10000					
2	5/30/96	Warmup	49	10000	0	10000	0.9	124	270	146
		Overrun	1	10000	0	10000	0.9	123	271	148
		Slow outer race speed increase	23	10000	10000	0	0.9	112	140	28
		Lockup, outer race drive, inner race none	5	10200	10200	0	0.9	111	139	28
3	5/31/96	Warmup	10	15000	15000	0	1.6	41	149	108
		Stop	2	0	0	0	1.6	120	121	1
		Accelerate inner race to 10,000 rpm	2	10000	0	10000	1.6	41	166	125
		Hold at 10,000 rpm overrun	6	10000	0	10000	2.7	41	250	209
		Increase speed	0.5	13500		13500	2.7	41	293	252
		Shutdown, oil temp high & not stabilizing				0	2.7			0

				Speeds: (rpm)			Flow (gpm)	Temp (°F)		
Run Nr	Date	Events	Duration Minutes	Notch Plate (I/R)	Pocket Plate (O/R)	Differ-ence	Oil In	Oil In	Oil Out	Delta
4	7/10/96	Check run, ok at 10,000 rpm	10	10000	0	10000	2.7			0
5	7/12/96	Warmup	10	3000	0	3000	2.7	170	161	-9
		Accelerate inner race to 10,000 rpm	0.5	10000	0	10000	2.7	165	191	26
		Overrun at 10,000 rpm	9	10000	0	10000	2.64	180	265	85
		Slowly increase outer race spd to engage	30	10000	10000	0	2.75	180	196	16
		Drive with outer race	0.5	10200	10200	0	2.75	180	195	15
6	7/16/96	Warmup	15	10000	0	10000	2.61	131	219	88
		Flow opt. 1.9 to 3.2 gpm, 2.5 gpm best	33	10000	0	10000	2.5	137	226	89
		11,000 rpm overrun	2	11000	0	11000	2.51	137	232	95
		12,000 rpm overrun	8	12000	0	12000	2.51	149	251	102
		13,000 rpm overrun	3	13000	0	13000	2.5	153	264	111
7	7/18/96	10,000 rpm overrun	58	10000	0	10000	2.67	112	168	56
		End of overrun		10000	0	10000	2.63	162	250	88
8	7/19/96	15,000 rpm overrun start	0.5	15000	0	15000	4.4	171	342	171
		Middle of run	4	15000	0	15000	3.9	178	356	178
		End of run	2	15000	0	15000	3	177	361	184
9	8/5/96	Warmup	8	20000	20000	0	2.56	128	156	28
		Slow outer race to 8,000 rpm	15	20000	8000	12000	2.58	178	273	95
10	10/23/96	Warmup (10,000 rpm overrun)	8	10000	0	10000	2.73	168	204	36
		Accelerate inner race to 15,000 rpm	18	15000	0	15000	2.76	159	248	89
		Overrun at 15,000 rpm	3	15000	0	15000	2.73	159	256	97
		Accelerate outer race to 15,000 & lock up	2	15400	15400	0	2.73	165	201	36
		Decelerate outerace to stop	1	15000	0	15000	2.7	162	181	19
		Overrun at 15,000 rpm	6	15000	0	15000	2.74	159	258	99
		Increase outer race speed to 10,000 rpm	2	15000	10000	5000	2.72	165	223	58
		Outer race speed to 4,000 rpm & hold	4	15000	4000	11000	2.72	168	296	128
		Slowly accel outer race and lock up	20	15100	15100	0	2.77	159	179	20
11	10/26/96	Overrun at 18,000 rpm	?	18000	0	18000				0
						0				0
12	10/28/96	Warmup	--	15000	10000	5000	2.58	147	186	39
		Increase inner race spd to 20,000 rpm	5	20000	10000	10000	2.64	156	252	96
		Increase outer race spd to lock up	1	20400	20400	0	2.63	162	202	40
		Stop outer race	1	20000	0	20000	2.57	166	314	148
		Hold at 20,000 overrun	5	20000	0	20000	2.67	200	374	174
		Increase outer race spd to 5,000 rpm	0.3	20000	5000	15000	2.65	200	407	207
		Increase outer race spd to 8000 rpm	0.1	20000	8000	12000	2.65	200	393	193
13	10/30/96	Overrun at 15,000 rpm	?	15000	0	15000				0
						0				0
14	11/1/96	Accelerate inner race to 15,000 rpm	8	15000	0	15000	2.6	88	176	88
		Overrun	30	15000	0	15000	2.7	116	239	123
		Changed oil temp controls	10	15000	0	15000	2.59	134	248	114

				Speeds: (rpm)			Flow (gpm)	Temp (°F)		
Run Nr	Date	Events	Duration Minutes	Notch Plate (I/R)	Pocket Plate (O/R)	Differ-ence	Oil In	Oil In	Oil Out	Delta
		Overrun with different oil temp settings	6	15000	0	15000	2.57	140	251	111
		Overrun with different oil temp settings	?	15000	0	15000	2.6	152	261	109
		Overrun with different oil temp settings	7	15000	0	15000	2.7	160	274	114
		Overrun with different oil temp settings	2	15000	0	15000	2.7	178	276	98
15						0				0
	11/1/96	Accelerate inner race to 20,000 rpm	2.5	20000	0	20000	2.78	174	297	123
		Overrun at 20,000	9	20000	0	20000	2.77	161	344	183
		Stop & check oil temps	15	0	0	0	1.98	72.5	131	58.5
16						0				0
	11/4/96	Slowly accel inner race to 17,500	35	17500	0	17500	2.32	121.7	149	27.3
		Stop & check oil temps	10	0	0	0	1.72	88	135	47
		Accel both races to 15,000	5	15000	15000	0	2.33	98	120	22
		Outer race to 17,500 & hold	5	17500	15000	2500	2.31	105	133	28
		Stop both races and stabilize	8	0	0	0	1.7	77	121	44
		Accel both races to 15,000	4	15000	15000	0	2.27	89	109	20
		Accel inner race to 17,500	0.5	17500	15000	2500	2.28	90	114	24
		Maintain speeds	3	17500	15000	2500	2.26	99	126	27
		Increase inner race speed to 20,000	0.1	20000	15000	5000	2.26	102	140	38
		Maintain speeds	3	20000	15000	5000	2.23	110	160	50
17						0				0
	11/5/96	Accelerate inner race to 22,000 rpm	2.5	22000	0	22000	2.68	117	263	146
		Maintain speed	3	22000	0	22000	2.66	140	322	182
		Accelerate outer race to lock up	2	22300	22300	0	2.64	155	207	52
		Stop both races	1	0	0	0	2.57	147	195	48
		Accelerate both races	2	23000	15700	7300	2.7	150	247	97
		Hold speeds	3	23000	15700	7300	2.69	150	233	83
		Decelerate to 10,000 rpm	1	10000	10000	0	2.67	145	179	34
		Increase speeds	0.5	23000	15000	8000	2.75	145	216	71
		Increase outer race speed to lock up	0.2	23200	23200	0	2.76	143	190	47
		Stop & check oil temps	1	0	0	0	2.27	129	181	52
18	11/6/96	Overrun to warmup	–			0				0
		Accelerate to initial speeds	4	17500	10000	7500	2.69	129	187	58
		Hold speeds	10	17500	10000	7500	2.74	168	236	68
		Increase speeds	2	20000	15000	5000	2.73	175	219	44
		Increase inner race speed	1	23000	15000	8000	2.76	175	245	70
		Increase speeds & lock up	1	25500	25500	0	2.72	170	219	49
19	11/21/96	Warmup, inc inner race spd to 15,000 rpm	12	15000	0	15000				#N/A
		Overrun	27	15000	0	15000				#N/A
20	11/26/96	Increase speed to 15,000 rpm	12	15000	0	15000		151		#N/A
		Overrun	7	15000	0	15000		166		#N/A
21	11/26/96	Increase speed to 15,000 rpm	27	15000	0	15000		166		#N/A

				Speeds: (rpm)			Flow (gpm)	Temp (°F)		
Run Nr	Date	Events	Duration Minutes	Notch Plate (I/R)	Pocket Plate (O/R)	Difference	Oil In	Oil In	Oil Out	Delta
		Overrun	30	15000	0	15000		166		#N/A
22	11/26/97	Increase speed to 10,000 rpm	4	10000	0	10000		99		#N/A
		Overrun	7	10000	0	10000		99		#N/A
23	11/26/96	Increase speed to 15,000 rpm	20	15000	0	15000		126		#N/A
						0				#N/A
24	12/4/96	Increase speed to 15,000 rpm	16	15000	0	15000		135		#N/A
		Overrun	9	15000	0	15000				#N/A
25	12/4/96	Increase speed to 15,000 rpm	14	15000	0	15000				#N/A
						0				#N/A
26	12/9/96	Increase speed to 20,000 rpm	17	20000	0	20000	2.78	200	331	131
		Overrun	20	20000	0	20000	2.68	223	371	148
27	12/11/96	Increase speed to 20,000 rpm	50	20000	0	20000	2.68	225	368	143
		Hold speed	3	20000	0	20000	2.7	226	382	156
		Reduce spd to 17,500 & hold	4	17500	0	17500	2.72	227	364	137
		Increase outer race speed to 9,500	2	17500	9500	8000	2.73	235	257	22
		Hold speed	2	17500	9500	8000	2.77	223	240	17
		Stop	2	0	0	0	0.1	212	262	50

I/R – Inner Race; O/R – Outer Race

First test was a spin up of the MD in locked mode to 10,000 rpm. Noise, vibration and temperatures appeared well within safe limits. This was followed by overrunning at 10,000 rpm with the outer race locked to ground. The upper oil outlet temperature appeared high and overrunning torque also measured unexpectedly high. Testing was stopped to determine the cause. On inspection, it was determined that the MD inner race shaft to test fixture upper adapter clearance was inadequate based on visual evidence of rubbing and high temperatures. The fixture adapter was relieved to provide additional clearance and the MD shaft was polished without disassembly. The MD oil feed hole was increased to approximately 0.12" diameter at this time also, providing oil supply up to about 3 gpm at maximum lubricant oil delivery pressure. The fixture was reassembled into the test rig.

Second test was an overrunning stair step. The inner race was brought up to 10,000 rpm, overrunning, with the outer race locked. The outer race speed was increased from 0 to 10,250 rpm in about 500 rpm increments over a period of 2 hour. The speed increase was not particularly linear with time, with the last 15 minutes of this test being done at overrunning differential speeds of less than 400 rpm. Engagement of the outer race to the inner race at 10,000 rpm was successful; evidenced by the inner race being driven to 10,250 rpm by the outer race. On completion of this test, the MD was disassembled and inspected. Some wear was evident on the struts and notch plate. Thickness of the struts was reduced by 0.002 inches on the top surface of the

two protrusions of each strut. Two corresponding wear rings were evident on the notch plate; these were measured as approximately 0.0006" deep. The MD was reassembled with new struts, springs and counterweights and reinstalled in the test rig.

Third test was a locked mode spin up of the MD to 15,000 rpm. This was accomplished without difficulty after modifying the oil supply tube mounting slightly to change its resonant frequency.

Fifth test was an attempt to overrun at 15,000 rpm. The outer race was locked and the inner race speed was increased incrementally. The test was terminated at 13,500 rpm due to excessive upper outlet oil temperature. The lubricant temperature at this outlet had reached 300 F with no sign that it would stabilize at or near this temperature. The MD was disassembled and examined. No high friction areas or areas of obvious overheating were discovered. A conclusion was reached that the oil overheating was due to oil shear and/or churning. A series of changes was determined to reduce the amount of oil shearing:

Increase the ID of the oil retention washer to 4.48 inches. This provides a sufficient oil outflow barrier to insure that the strut area remains flooded without additional unnecessary oil shear area.

Reduce the radial size of the axial load contact on the back of the notch plate to half its original dimension and increase the axial relief distance in the vicinity of the ring to 0.05 inches. This minimizes the low clearance oil shear area.

Increase the number of relief slots in the axial load contact ring on the back of the notch plate from 4 to 8 and add a 10 degree trailing edge ramp. The increased number of slots will promote oil flow for cooling and the trailing edge ramps will increase the likelihood of formation of a lubricating oil wedge under all conditions.

Drill six 1/4 inch diameter oil drain holes in the face of the pocket plate to reduce churning of trapped oil between the notch plate and the bearing housing portion of the pocket plate.

An additional change was being made, unrelated to the high lubricant temperature problem. A new set of counterweights was being fabricated to reduce the axial force on the struts by about 50%. The weight reduction was achieved by removing non-essential portions of the cam without changing its strut contacting profile. This was intended to reduce the above mentioned strut wear. This wear was believed a result of excessive counterweight force at low differential but high absolute speeds.

High speed testing was then continued with Test #7. Modifications to prototypes

were completed on two prototypes. The parts were forwarded to BHTI and subsequently were evaluated according to the test plan. Overrunning and MD engagement were confirmed at 10,000 rpm and a successful overrun, engage series was completed at 15,000 rpm. Based on these results, BHTI attempted a 20,000 rpm test sequence.

At 20,000 rpm, oil temperature was high and did not stabilize within reasonable limits, so the experiment was terminated and BHTI reported the results. Since the previous modifications substantially improved the oil heating characteristics of the MD at high overrunning speed, we decided to undertake further modifications to reduce the remaining oil shear areas. The changes were as follows.

Increase the diameter of the oil drain holes added to the pocket plate from 1/4 inch to 3/8 inches to insure that the inner portions of the MD remain free of liquid at the high oil delivery rates required to achieve part cooling during high speed overrunning.

Relieve the surface of the pocket plate between the three pockets. Mill away the plate surface to a depth of approximately 0.05 inches between the pocket, leaving each pocket formed in the surface of a small pedestal.

Increase the diameter of the turned face relief on the pocket plate to minimize the unrelieved radial distance at the pocket pedestal.

Decrease the radial dimension of the axial load bearing ring on the outer side of the notch plate to 1/8 inch, and increase the axial relief at this feature to 0.06 inches.

Drawings with these changes were prepared and three prototypes were sent to a local machine shop to accomplish the changes. Copies of the drawings are attached as Appendix C.

Tests #10 through #18 were carried out by BHTI on fully modified samples as described above. Initial runs were designed to explore the improvement in lubricant temperature rise, if any. These seven test runs were conducted under cold to warm oil inlet temperature conditions. Oil temperature stabilized at overrunning speeds up to 22,000 rpm. During this test series, BHTI determined that the reliable upper speed limit of the test rig was 20,000 rpm. Sustained operation at speeds significantly higher than this limit were judged likely to result in premature bearing failures.

A final pair of test runs were conducted to check the MD's oil heating behavior under hot oil inlet conditions. Extended overrunning at 20,000 rpm differential speed, with an inlet oil temperature of 225°F resulted in a stable temperature rise across the MD of

143°F. This temperature increase was judged conditionally acceptable.

Part wear, especially the upper surface of the struts continued higher than desirable. The continued efforts to decrease the strut force exerted by the counter weights again resulted in decreased strut wear; however, even with test runs of only a few hours, there was visible strut polishing and wear. Additional effort is required to improve the surface finish of the strut opposing surfaces and of the struts them selves to allow very small gap operation without contact and resulting wear. Further, strut surface hardness needs to be increased to improve the strut's resistance to occasional contact with mating parts. These modifications need to be formulated, prototypes modified and finally the prototypes tested to verify acceptable wear levels before this design can be considered ready to develop into a production version.

5. Life Cycle Testing

Although the primary objective of this program was to investigate the high speed overrunning characteristics of the MD, verification that the design being evaluated was suitable for carrying the required torque during lock up was required. To perform this verification, the following test plan was devised.

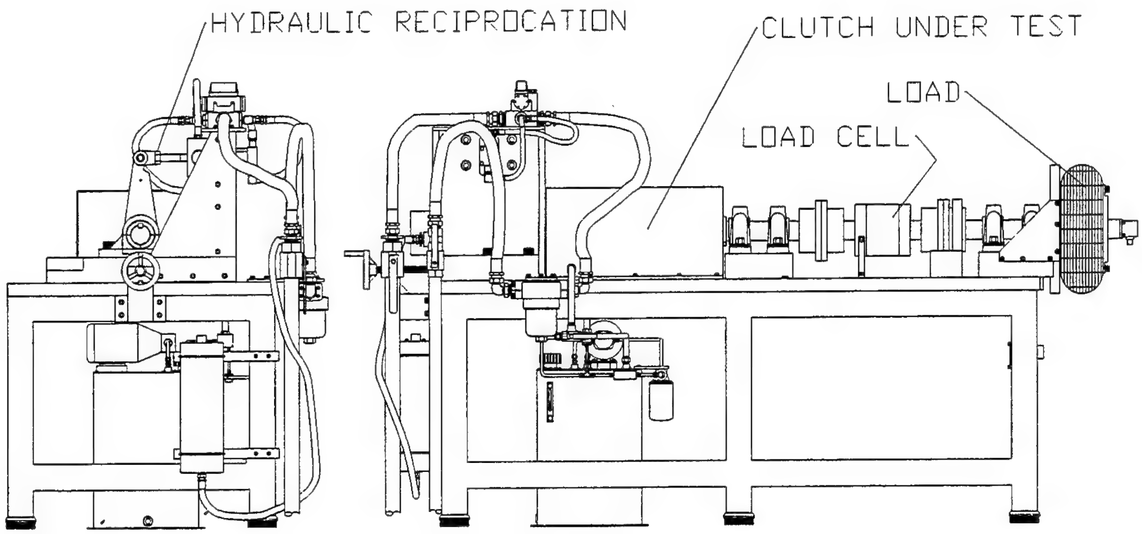
Torque requirements are as set forth below:

Maximum Continuous Torque	875 ft-lb
Limit Torque (200% max)	1,750 ft-lb
Yield Torque (230% max)	2,008 ft-lb
Ultimate Torque (300% max)	2,625 ft-lb

The objective of the load cycle tests is to provide confidence in the proposed clutch design by demonstration of the following.

- Repeated loading to the Maximum Continuous Torque with no damage or wear.
- Load cycle life in excess of 500,000 cycles at the Limit Torque load.
- Linear torque/angular deflection characteristics to at least the Yield Torque value.
- Ultimate strength in excess of the Ultimate Torque value.

The test specimen consisted of a pocket plate (P/N 100869C1), a notch plate (P/N 100865C1), 3 cams (P/N 100871), 3 struts (P/N 200213), 3 springs (P/N 100545), 2 lock washers (P/N 200216), retainer (P/N200214, and Smalley retaining ring WH-590. This test assembly was designed to operate in a high speed overrunning fixture; therefore, the load cycle fixture, including appropriate



adapters was designed to accommodate the test assembly. The load cycle test fixture is shown in Figure 12.

Figure 12

The tests were conducted in the load cycle fixture shown in Figure 12. The tester was located at a subcontractor's facility. The input (notch plate 100865) shaft is driven by a hydraulic cylinder acting on a lever arm which is attached to the shaft. The shaft is driven alternately clockwise and counterclockwise and this causes the MD to drive the output (pocket plate P/N 100869) in one direction but discontinuously. The output shaft is connected to a disk brake through a torsional load cell. The disk brake serves as a low power, high torque dynamometer and loads the MD output shaft to a selected level. The test specimen is immersed in lubricant which remains at approximately room temperature. The test lubricant was commercial automatic transmission fluid.

The following test parameters will be monitored and recorded at the standard system interval during all testing:

Time of Day
Total Test Time
Input Shaft Cycle Rate
Output Shaft Torque

The load cycle performance tests shall be in accordance with Table 2

Step	Rate (cycles/sec)	Load (ft-lb)	Run Time (cycles)
1	1	400	100
2	2	400	100
3	2	875	100
4	3	875	20,000
5	3	875	250,000

Table 2

The test steps in this table describe a testing sequence to evaluate the clutch at the maximum continuous torque. Steps 1 through 4 are a planned set of initial tests to verify that the test rig and sample are performing properly. Step 5 is a realistic test to verify that the clutch has nominally adequate fatigue characteristics. This test sequence is the primary verification that the size range of the prototypes is appropriate for the intended applications.

On completion of the above, a second test assembly shall be installed in the test fixture and tested to failure according to Table 3.

Step	Rate (cycles/sec)	Load (ft-lb)	Run Time (cycles)
1	3 to 5	1,750	to failure

Table 3

This table specifies testing of a prototype clutch at the limit torque, a torque load twice the maximum projected load.

On completion of tests in Table 3, a third test assembly shall be installed in the test fixture and tested to failure or to the load capacity of the test fixture according to Table 4.

Step	Rate (cycles/sec)	Load (ft-lb)	Run Time (cycles)
1	.025	to failure	1

Table 4

This table specifies a one time loading of the clutch to verify that torsional stiffness is as expected. The torque vs twist angle from one end of the clutch to the other was expected to be essentially linear within the range of normal use.

Several design approaches were considered for the test system to carry out this test plan; the tester to be built for this project was a modified unit built by a subcontractor.

Changes to the design were kept to a minimum, but included at least increasing the machine stroke and the length of the sample mounting enclosure. Specifications for the tester are:

Capacity - 1,800 ft-lb nominal
3,000 ft-lb dynamic
3,200 ft-lb ultimate
Speed - up to 8 cycles per second
Stroke - 0 to 14°

Initial Tests — Steps 1 through 4 of Table 2 were carried out during a project review. On completion of this sequence of tests, the sample clutch was removed from the test rig, disassembled and inspected. All parts were in the same condition as when the test started except that slight shininess on the load bearing strut ends was evident. Examination of the strut ends revealed slight surface asperity height reduction from repeated loading.

1X Fatigue — Step 5 of Table 2 was completed and the parts returned for inspection. No signs of distress were found during this inspection. Strut ends showed slight surface asperity reduction from compressive loading and a small amount of fretting corrosion was evident within the MD bearing housing. Based on life cycle tests of other similar automotive parts, these observations are believed to be normal for parts which have been subjected to a 250,000 cycle test. No changes or part modifications are planned.

2X Fatigue — 2X load fatigue testing was carried out. The initial test at this torque was terminated at 2,091 cycles due to failure of the small male spline on the test fixture. A new one piece replacement fixture was made and and heat treated for improved strength. Testing was restarted with a new specimen. The sample failed after 296,143 load cycles. Failure was a spiral fracture of the hollow shaft immediate adjacent to the small spline

connection on the notch plate part of the MD. The mating portion of the new fixture was also damaged as a result of the shaft fracture. This shaft and the small spline incorporated into it are sized to fit the BHTI high speed overrunning test rig. Examination of the MD revealed no evidence of distress, cracking or unusual material deformation. Based on this testing, it is likely that this design will meet the torque carrying portions of the development specification.

Life cycle test samples were inspected and cause of failure determined as described in above. The one-way clutch portions of the life cycle test samples was virtually unaffected during testing up to the point of failure of the shafting sized for the high speed overrunning rig.

No strength or life related modifications to the one-way clutch are planned since the load handling capability of the prototypes appeared to be adequate.

6. Recommendations/Future Work

Over the course of this program, development of the MD for high speed overrunning dealt primarily with the thermal problems associated with high speed shearing of the lubricant. Concern of component wear were not adequately addressed and represent additional work required to fully develop the MD for high speed use. The part wear observed during testing was by no means catastrophic; however, sufficient strut and spring wear was observed to suggest that this issue will require improvement to assure adequate service life for the MD. Several material surface treatment changes have been proposed including nitride hardening, carbo-nitriding and titanium nitride coating. Surface finish improvements on the hardened parts should also reduce initial wear. A program will be required to prepare parts intended for wear resistant behavior and to evaluate the parts by high speed testing with quantitative evaluation of wear behavior by gravimetric or other means. On completion of such an additional program, this clutch design would be provisionally suitable for application at the turbine shaft in a rotorcraft driveline.

7. Summary

During the course of this program, two MD type overrunning clutches for high speeds were designed. One of the designs was implemented as a set of prototype clutches for high speed overrun testing. A high speed test stand was utilized for performing overrunning and engagement tests at speeds up to 20,000 rpm. MD overrunning

clutches were tested at moderate speed, up to 10,000 rpm and substantial thermal problems associated with oil shear were encountered. The MD design was modified, the modified parts were tested, and by program end, clutches were tested in excess of 20,000 rpm without excessive lubricant temperatures. Some correctable wear was observed and remains as a clutch characteristic which needs further improvement. A load cycle tester with a special, long sample section was designed, built and then prototype clutches were fatigue tested to verify that the clutch design was suitable for carrying the specified power levels. Load cycled parts were subjected to approximately 300,000 cycles of twice normal load with the clutch locking elements showing no signs of distress. Adjacent torque carrying portions of the clutch did not allow longer testing; however, it is believed that the clutch strength and life was shown to be adequate.

References

[1] – TK Solver Plus, Universal Technical Systems, Rockford Illinois

[2] – Roark's Formulas for Stress and Strain, McGraw-Hill, Copyright 1989

Appendix A

High Speed Overrunning Test Plan

1.0 INTRODUCTION

As part of a Phase II SBIR contract to design, fabricate and evaluate an aircraft quality lightweight overrunning clutch for operation at speeds from 0 to 30,000 rpm, BHTI has been subcontracted by Epilogics, Inc. to conduct the clutch overrunning and engagement tests. The program is directed by the NASA Lewis Research Center, Cleveland, OH.

The objective of the overrunning and engagement tests is to provide confidence in the proposed clutch design by demonstration of the following:

- Overrunning at maximum oil-in temperature and worst case speed differential with no damage or wear.
- Successful engagement after overrunning simulating rotorcraft engine ramp rates.
- Successful engagements at cold temperatures simulating rotorcraft engine ramp rates.
- Fail-safe failure (no lock-up) during 30 minute loss-of lube operation.

2.0 TEST SPECIMEN

The test specimen shown in Figure 1 consists of a pocket plate (P/N 100869C1), a notch plate (P/N 100865C1), cam (P/N 100871, 3 pl), strut (P/N 200213, 3 pl), spring (3 pl), lock washer (P/N200216, 2 pl), retainer (P/N 200214), spacer (P/N 200313), and backing ring (P/N 100872) with Smalley retaining ring WH-590. The test assembly was designed as shown to operate in the existing test fixture assembly shown in Figure 2.

3.0 TEST APPARATUS AND INSTRUMENTATION

The tests will be conducted in cell 103 using the high speed overrunning test fixture shown in Figure 2. The input (Pocket Plate 100869C1) and output (Notch Plate 100865C1) shafts are independently driven by variable speed motors through speed increaser gearboxes at variable speeds from 0 to 30,000 rpm. Operation at speeds above 20,000 rpm will be permitted only if safe and stable vibration levels are demonstrated and maintainable. The drive motors only provide the torque necessary to overcome the frictional, windage and churning losses of the clutch assembly and the required acceleration rates. No torsional loading devices (e.g. dynamometers) are required for these tests. Additionally the input drive motor is capable of accelerating the pocket plate at 2600 rpm/sec (simulates engine start-up) up to 20,000 rpm. Separate lube systems supply oil to the test specimen and speed increaser gearboxes. The test specimen lube system is capable of supplying lubrication at -40 to +230° F with variable pressure from 0 to 100 psig. The test lubricant will be DOD-L-85734. An insulated refrigeration box will be constructed around the test specimen for the cold engagement tests.

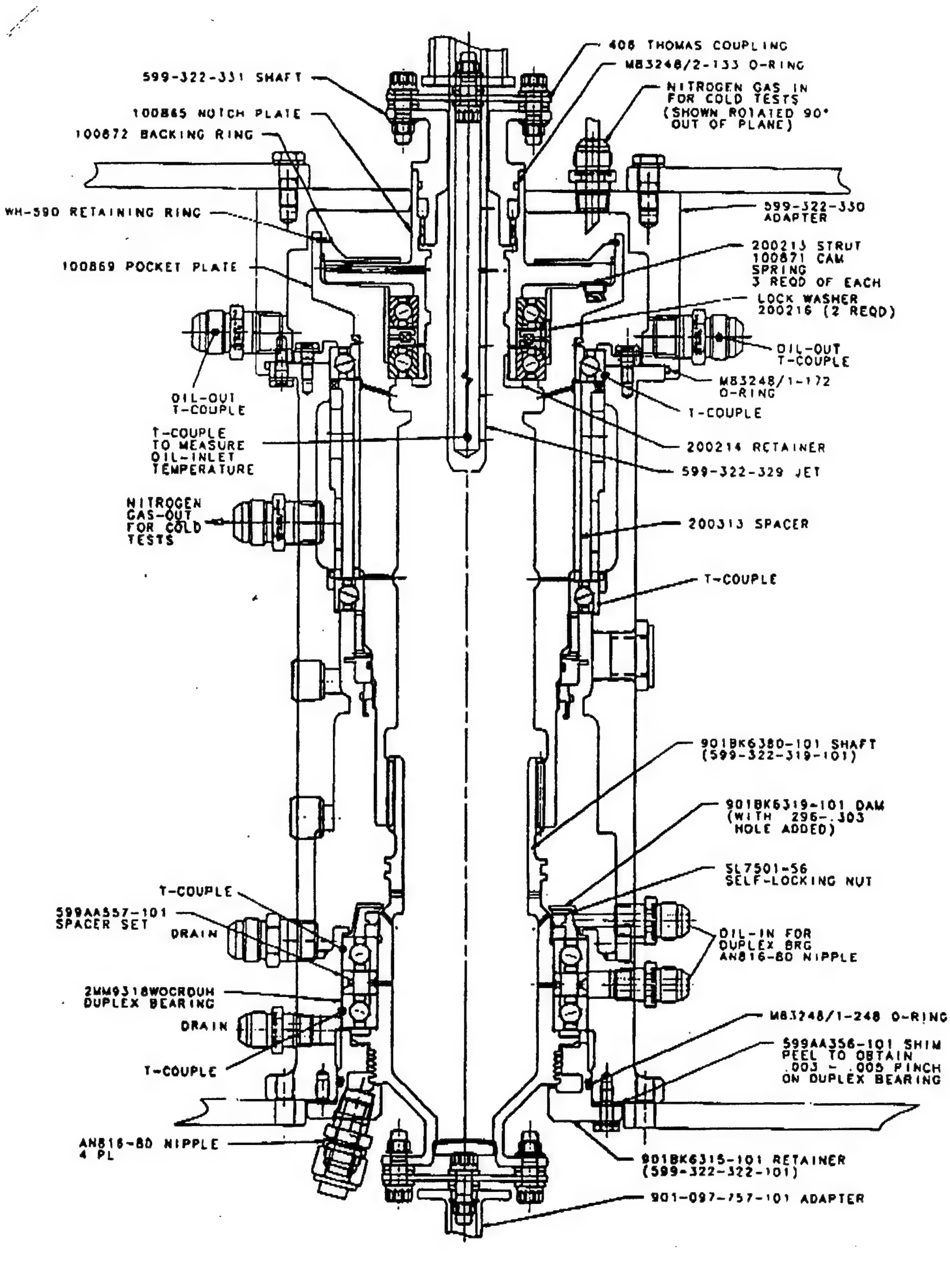


Figure 1: Mechanical Diode Clutch Assembly Installation

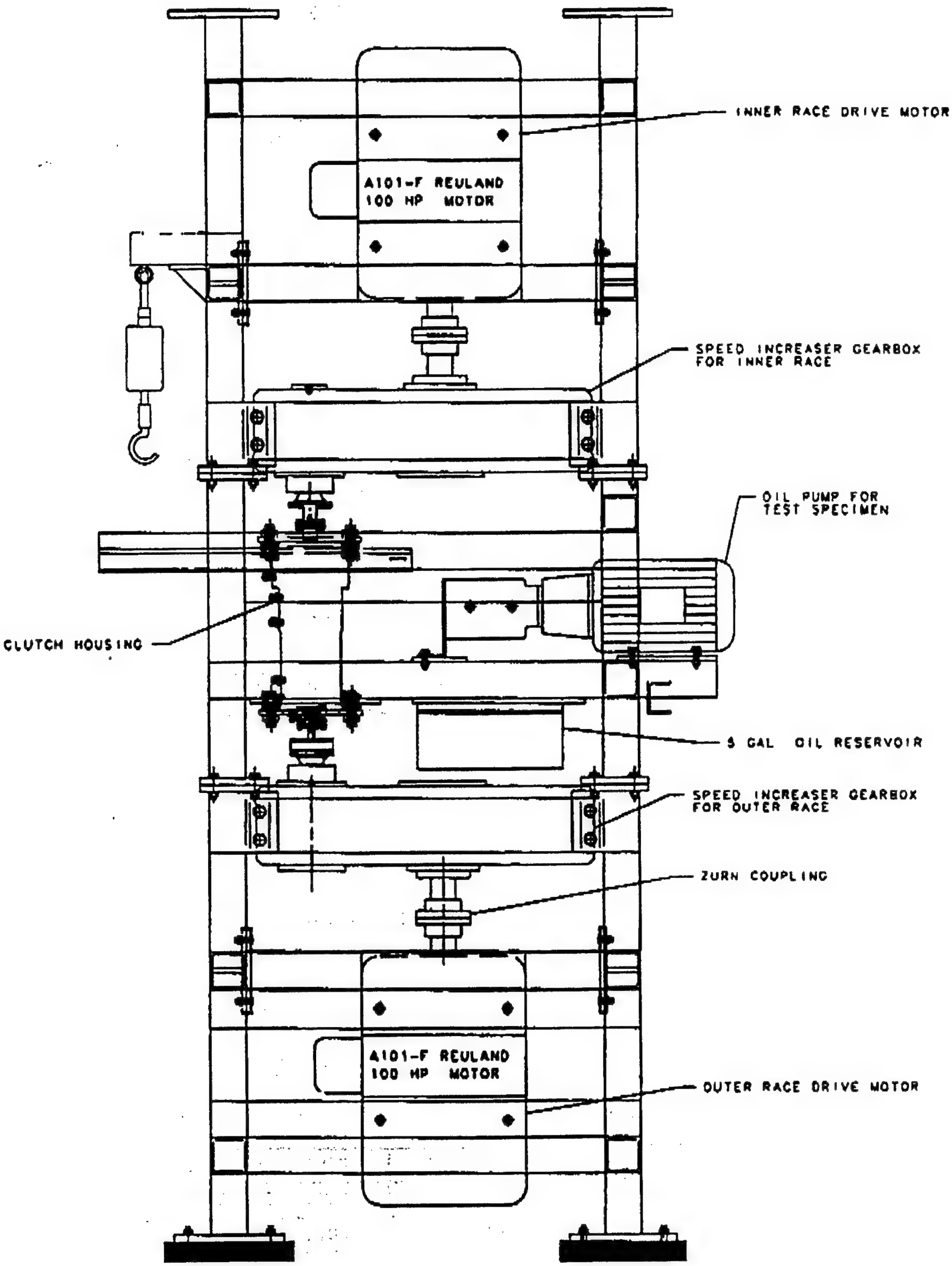


Figure 2: Clutch Overrunning Test Stand

The following test parameters will be monitored and recorded at 2 second intervals during all testing:

Time of Day
Total Test Time
Test Cell Ambient Temperature
Input Shaft rpm and Rate of Acceleration
Input Shaft Torque
Output Shaft rpm
Oil Inlet Temperature (measured at Jet Discharge as shown in Figure 1)
Oil Outlet Temperature (2 places as shown in Figure 1)
Oil Inlet Flow Rate and Pressure (measured at Jet Inlet)
Support Bearing Temperatures (4 places as shown in Figure 1)

4.0 OIL FLOW OPTIMIZATION/SHAKEDOWN TESTS

The oil flow optimization/shakedown tests shall be conducted in accordance with Table I using DOD-L-85734 oil.

With the initial oil inlet pressure held at 60 ± 5 psig at 160° ± 10° F inlet temperature, conduct the five differential speed sweep tests of Table I. Set the notch plate speed as specified for each step, then increase the pocket plate speed from 0 to the specified speed at a constant rate of increase over the 0.5 hour run time. When the maximum pocket plate speed is reached, shut off the notch plate drive motor, and verify clutch engagement. During each step, the oil flow rate through the clutch assembly shall be increased as required by increasing the oil inlet pressure in order not to exceed 230° F oil outlet temperature while maintaining the oil inlet temperature at 160° ± 10° F.

If the oil-out temperature exceeds 230° F, the test shall be suspended and the clutch disassembled for inspection. If no wear or damage has occured, the clutch will be reassembled and the test step repeated with a 10° F increase in the oil-out temperature allowed. Repeat this process as necessary to complete the test step. All testing will be suspended if excessive wear or damage is observed after any step.

Steps 4 and 5 shall only be conducted if stable operation of the test stand can be maintained.

After each test step the clutch assembly shall be disassembled, visually inspected and photographed. Dimensional inspections shall be made to document any wear on the notch plate, pocket plate, cams, or struts.

Table I: Oil Flow Optimization Tests

STEP	POCKET PLATE SPEED (RPM)	NOTCH PLATE SPEED (RPM)	RUN TIME (HR)
1	0 - 10,200	10,000	0.5
2	0 - 15,200	15,000	0.5
3	0 - 20,200	20,000	0.5
4	0 - 25,200	25,000	0.5
5	0 - 30,200	30,000	0.5

5.0 HOT OVERRUNNING AND ENGAGEMENT TESTS

The hot overrunning and engagement tests shall be conducted in accordance with Table II using DOD-L-85734 oil at 230° ± 5°F. The optimum oil inlet pressure ± 5 psig, as determined in the oil flow optimization tests, shall be held throughout each test step. The same components used in the oil flow optimization tests will be used for this test if acceptable to Drive System Design Engineering.

Steps 3 and 4 shall only be conducted if stable operation of the test stand can be maintained.

With the notch plate set at the speed specified for each step, increase the pocket plate speed from 0 to the intermediate speed specified in Table II at a constant rate of increase over a .5 hour run time, then increase the pocket plate speed from the intermediate speed to the final speed at a rate of 3000 ± 600 rpm/sec or at the maximum acceleration rate capability of the pocket plate drive motor if less than 2400 rpm/sec. When the maximum pocket plate speed for each step is reached, shut off the notch plate drive motor, and verify clutch engagement.

After each test step the clutch assembly shall be disassembled, visually inspected and photographed. Dimensional inspections shall be made to document any wear on the notch plate, cams, and struts.

Table II: Hot Overrunning and Engagement Tests

STEP	POCKET PLATE SPEED (RPM)	NOTCH PLATE SPEED (RPM)	RUN TIME (HR)
1	0 - 13,000 - 15,200	15,000	0.5
2	0 - 18,000 - 20,200	20,000	0.5
3	0 - 23,000 - 25,200	25,000	0.5
4	0 - 28,000 - 30,200	30,000	0.5

6.0 COLD ENGAGEMENT TESTS

The cold engagement tests shall be conducted in accordance with Table III using DOD-L-85734 oil at the oil reservoir temperatures specified. The same components used in the hot overrunning and engagement tests will be used for these tests if acceptable to Drive System Design Engineering.

The 1st engine start tests, steps 1 & 2, shall be performed per the following procedure (Step 2 shall only be conducted if stable operation of the test stand can be maintained):

Bring the components and oil reservoir to the specified initial temperatures ± 5° F.

Accelerate the pocket plate to the specified test speed at a uniform rate of 300 ± 50 rpm/sec. . Verify notch plate speed matches pocket plate speed. Notch plate motor shall remain off during entire run.

Oil pressure shall be provided to the clutch 5 to 10 seconds after start of pocket plate rotation. Oil pressure shall be increased uniformly to 100 psig as pocket plate reaches specified speed.

The 2nd engine start tests, steps 3 through 12, shall be performed per the following procedure: (Steps 8 through 12 shall only be conducted if stable operation of the test stand can be maintained.)

Bring the components and oil reservoir to the specified initial temperatures ± 5° F.

Accelerate the notch plate to the specified test speed at a uniform rate of 300 ± 50 rpm/sec, then dwell at the specified speed. The total run time from start of the notch plate rotation to the end of the notch plate dwell time shall be 120 seconds.

Oil pressure shall be provided to the clutch 5 to 10 seconds after start of the notch plate rotation and shall increase uniformly to 100 ± 10 psig as the notch plate speed reaches the specified test speed. During the dwell time, the oil pressure shall decrease from 100 psig to the optimum oil inlet pressure ± 5psig, as determined in the oil flow optimization tests.

At the end of the notch plate dwell time, the pocket plate rotation shall be initiated and accelerated from 0 to the specified test speed plus 200 rpm in 40 seconds. The pocket plate acceleration shall be 1300 ± 250 rpm/sec at engagement. Verify engagement.

After each test step the clutch assembly shall be disassembled, visually inspected and photographed. Dimensional inspections shall be made to document any wear on the notch plate, pocket plate, cams, and struts.

TABLE III: Cold Temperature Engagement Tests

TEST STEP	CONDITION	INITIAL COMPONENT & RESERVOIR TEMPERATURE	TEST SPEED (RPM)
1	1st Engine Start	-40° F	20,000
2	1st Engine Start	-40° F	30,000
3	2nd Engine Start	0° F	20,000
4	2nd Engine Start	-10° F	20,000
5	2nd Engine Start	-20° F	20,000
6	2nd Engine Start	-30° F	20,000
7	2nd Engine Start	-40° F	20,000
8	2nd Engine Start	0° F	30,000
9	2nd Engine Start	-10° F	30,000
10	2nd Engine Start	-20° F	30,000
11	2nd Engine Start	-30° F	30,000
12	2nd Engine Start	-40° F	30,000

7.0 45 HOUR OVERRUNNING WEAR EVALUATION

The 45 hour overrunning wear evaluation test shall be conducted in accordance with Table IV using DOD-L-85734 oil at 200° ± 5°F. The optimum oil inlet pressure ± 5 psig, as determined in the oil flow optimization test, shall be held throughout each test step. A new set of clutch components shall be assembled for this test (notch plate, pocket plate, struts, springs and cams).

With the notch plate set at the speed specified for each step, increase the pocket plate speed from 0 to the intermediate speed specified in Table IV at a constant rate of increase over the 0.5 hour run time, then increase the pocket plate speed from the intermediate speed to the final speed at a rate of 3000 ± 600 rpm/sec or at the maximum acceleration rate capability of the pocket plate drive motor if less than 2400 rpm/sec. When the maximum pocket plate speed for each step is reached, shut off the notch plate drive motor, and verify clutch engagement. After each step, return the pocket plate to 0 rpm then set notch plate speed for the next step.

Repeat Table IV 30 times to accumulate 45 hours of run time. Test steps do not have to be run consecutively, e.g. the test may be stopped after step 1 then resumed starting step 2.

Step 3 shall only be conducted if stable operation of the test stand can be maintained. If step 3 can not be run, then repeat steps 1 and 2 a total of 45 times to accumulate 45 hours.

The clutch assembly shall be disassembled, visually inspected and photographed after each 9 hours of run time (5 disassemblies in 45 hours). Dimensional inspections shall be made to document any wear on the notch plate, pocket plate, cams, and struts.

Table IV: 45 Hour Overrunning Endurance Test

STEP	POCKET PLATE SPEED (RPM)	NOTCH PLATE SPEED (RPM)	RUN TIME (HR)
1	0 - 8,000 - 10,200	10,000	0.5
2	0 - 18,200 - 20,200	20,000	0.5
3	0 - 28,200 - 30,200	30,000	0.5

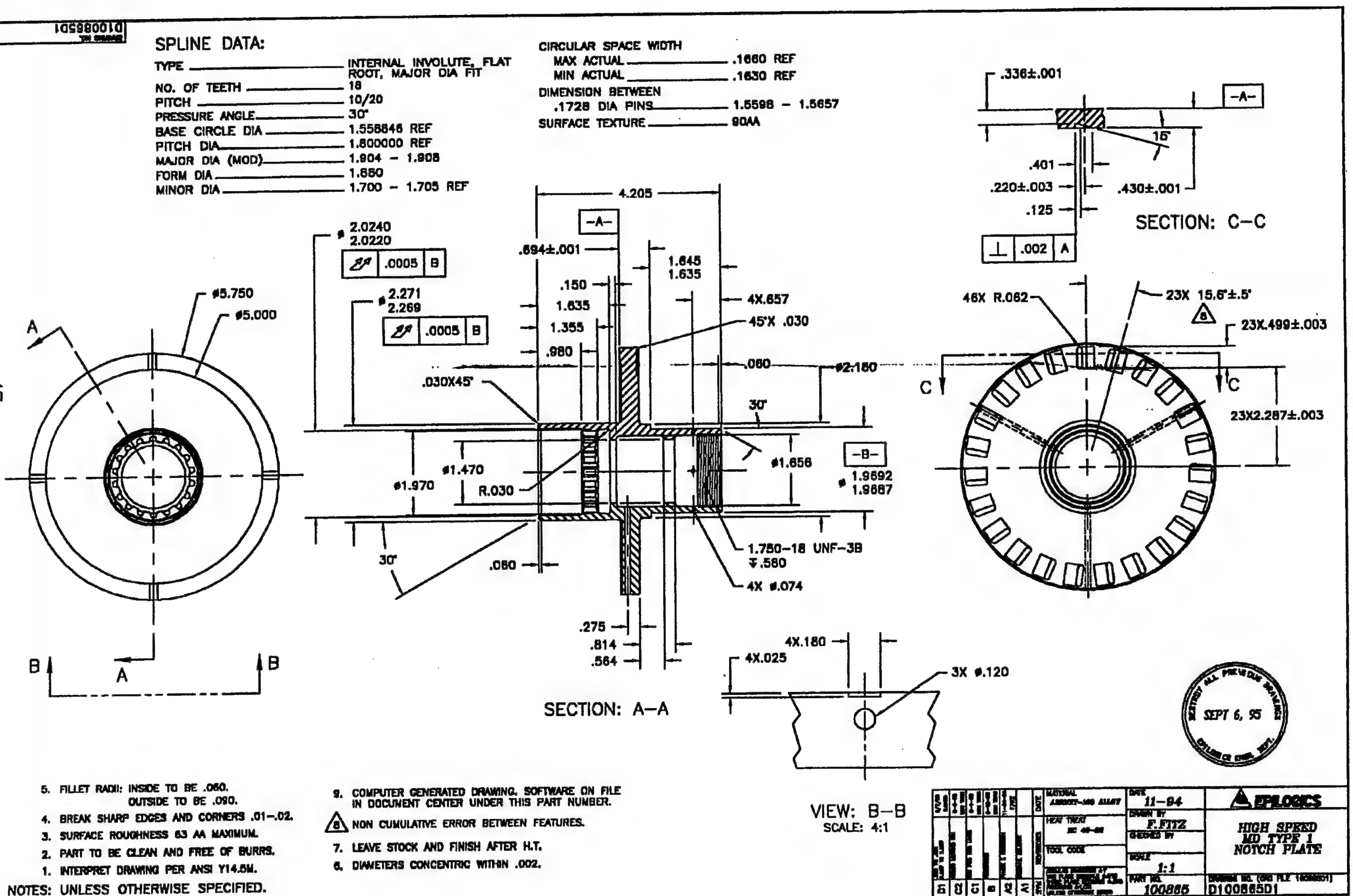
8. LOSS-OF-LUBE TEST

The loss-of-lube test shall be conducted using DOD-L-85734 oil.

Operate the clutch at the maximum stable clutch stand speed up to 30,000 rpm and with oil-in temp at 200° ± 10° F at the optimum oil inlet pressure ± 5 psig, as determined in the oil flow optimization test. The clutch shall be engaged with the pocket plate driving and the notch plate driving at 200 rpm slower than pocket plate. When the oil-out temperature stabilizes (no increase in temperature in 2 minute period), cut-off the clutch oil supply and reduce the pocket plate speed to 0 rpm. Run for one hour or until the clutch re-engages due to failure or until the notch plate drive motor fails to transmit torque.

Appendix B

Initial Version
High Speed Overrunning Test Prototype
Drawings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PITCH DIA — 1.800000 REF
MAJOR DIA (MOD) — 1.904 – 1.908
FORM DIA — 1.880
MINOR DIA — 1.700 – 1.705 REF
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DIMENSION BETWEEN .1728 DIA PINS — 1.5598 – 1.5657
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ø5.000
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B
ø 2.0240 2.0220
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-A-
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1.355
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.060
ø2.180
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.401
.220±.003
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.430±.001
.002 A
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8 NON CUMULATIVE ERROR BETWEEN FEATURES.
7. LEAVE STOCK AND FINISH AFTER H.T.
6. DIAMETERS CONCENTRIC WITHIN .002.
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F.FITZ
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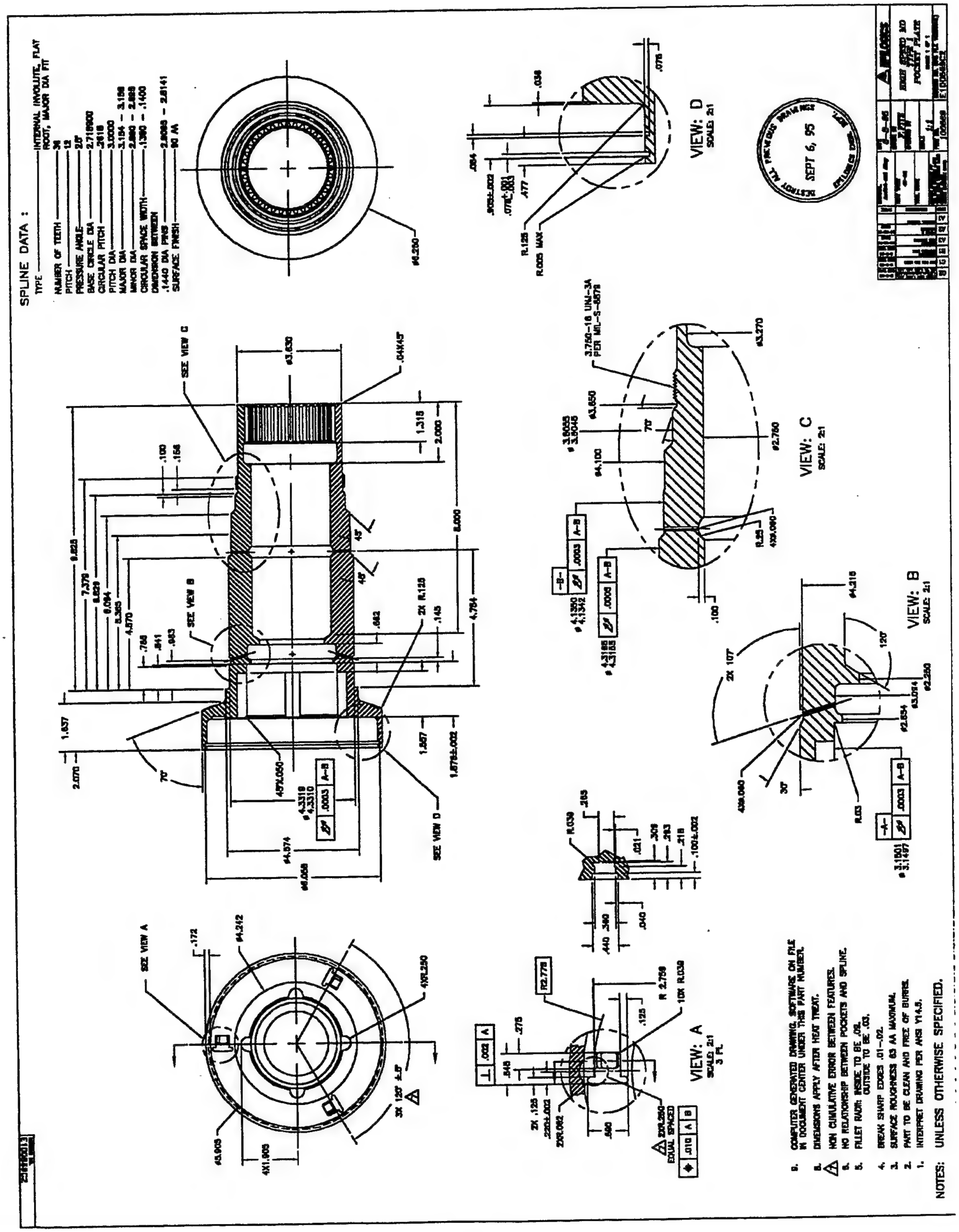
HIGH SPEED MD TYPE 1 NOTCH PLATE
D100865D1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SURFACE FINISH 32 AA
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3 PL
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VIEW: C
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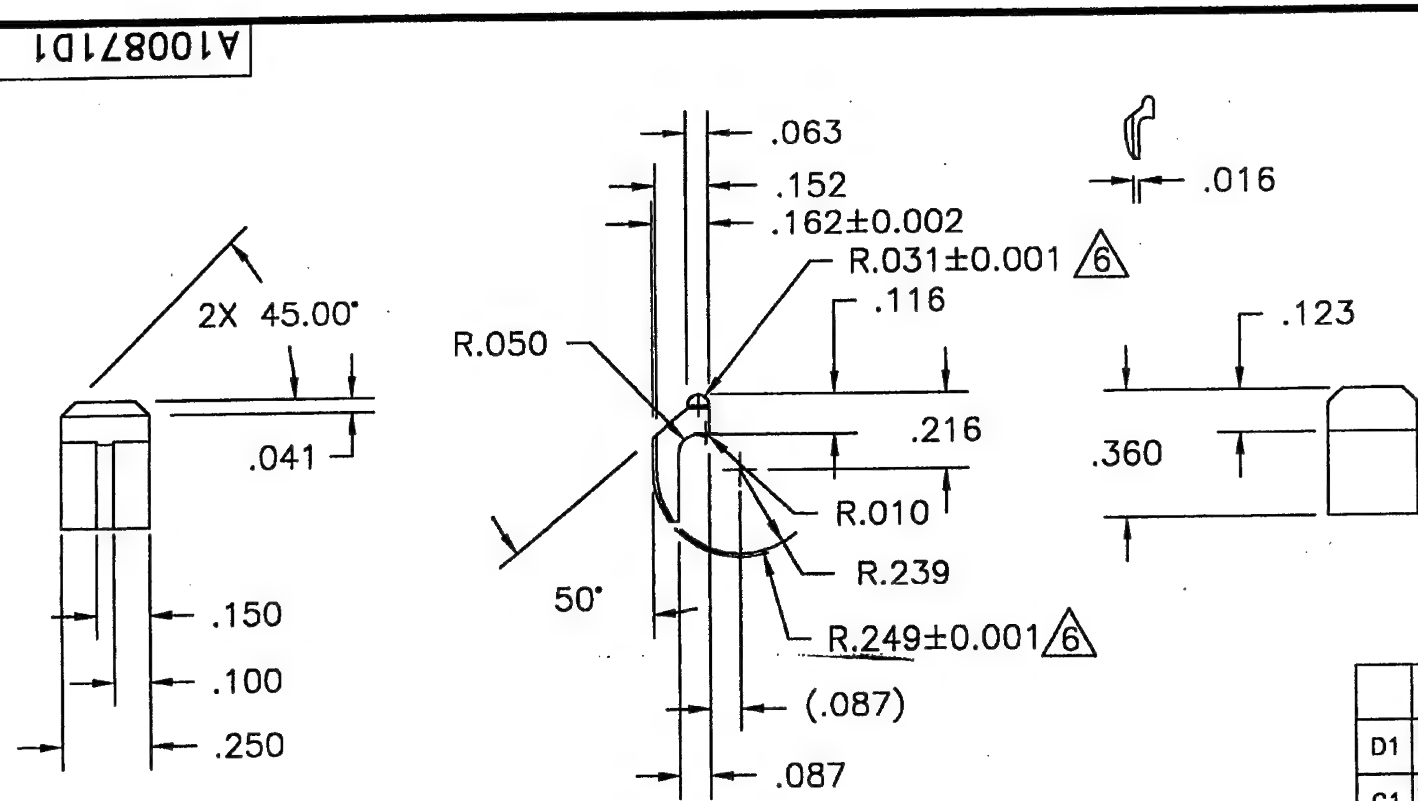

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B1	FORMATTED	6-12-95 DON TRAN
A2	PHASE 2 RELEASE	11-22-94 F.FITZ
A1	ORIGINAL RELEASE	

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	CHECKED BY	
ANGULAR DIMENSION ± 1° FRACTIONS ±1/32 TWO PLACE DECIMALS ±.010 THREE PLACE DECIMALS ±.005 UNLESS OTHERWISE NOTED	SCALE *1:1*	SHT 1 OF 1
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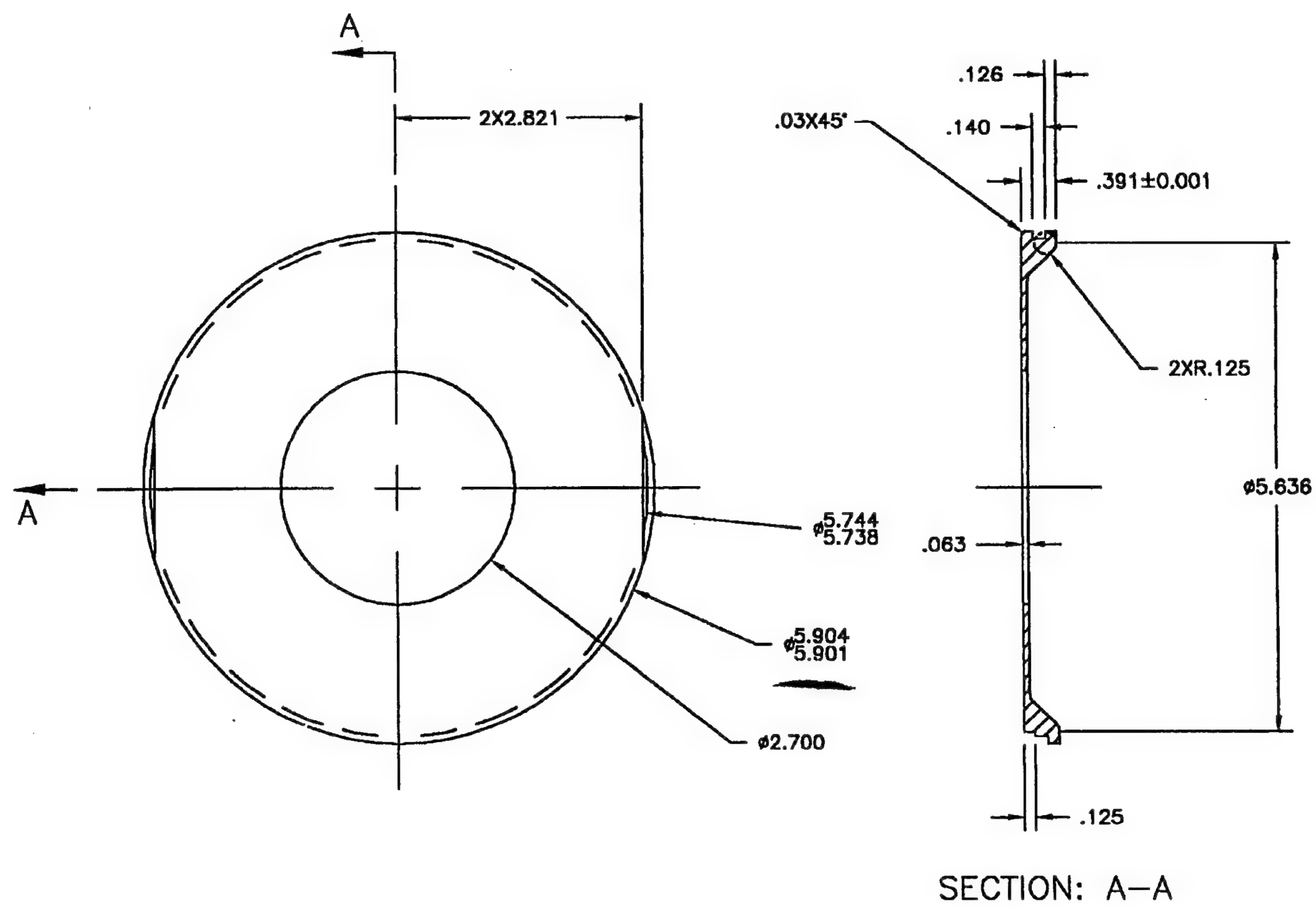

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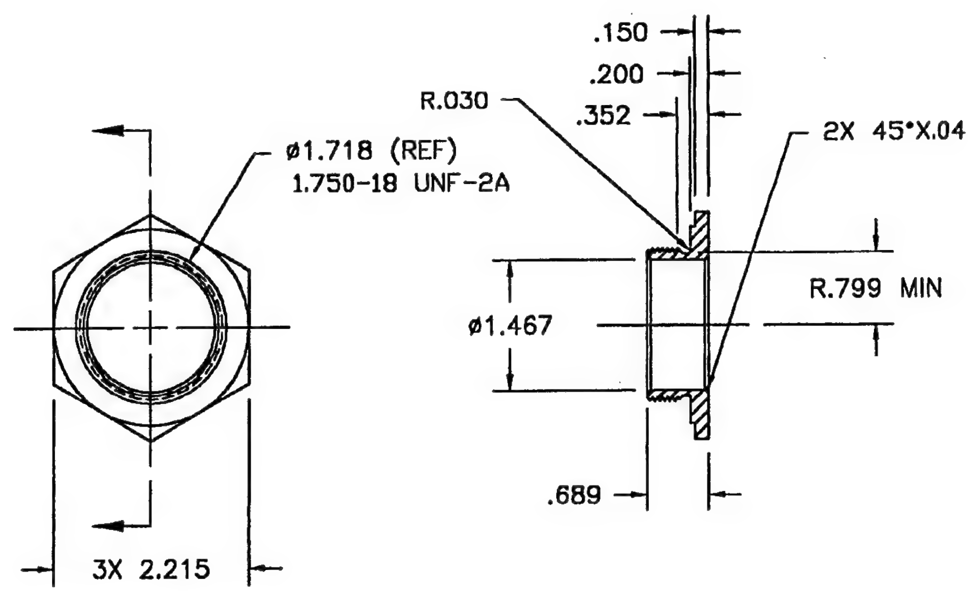

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JUNE 12, 95
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B1	FORMATTED	8-12-95 DOC TRAN
B1		

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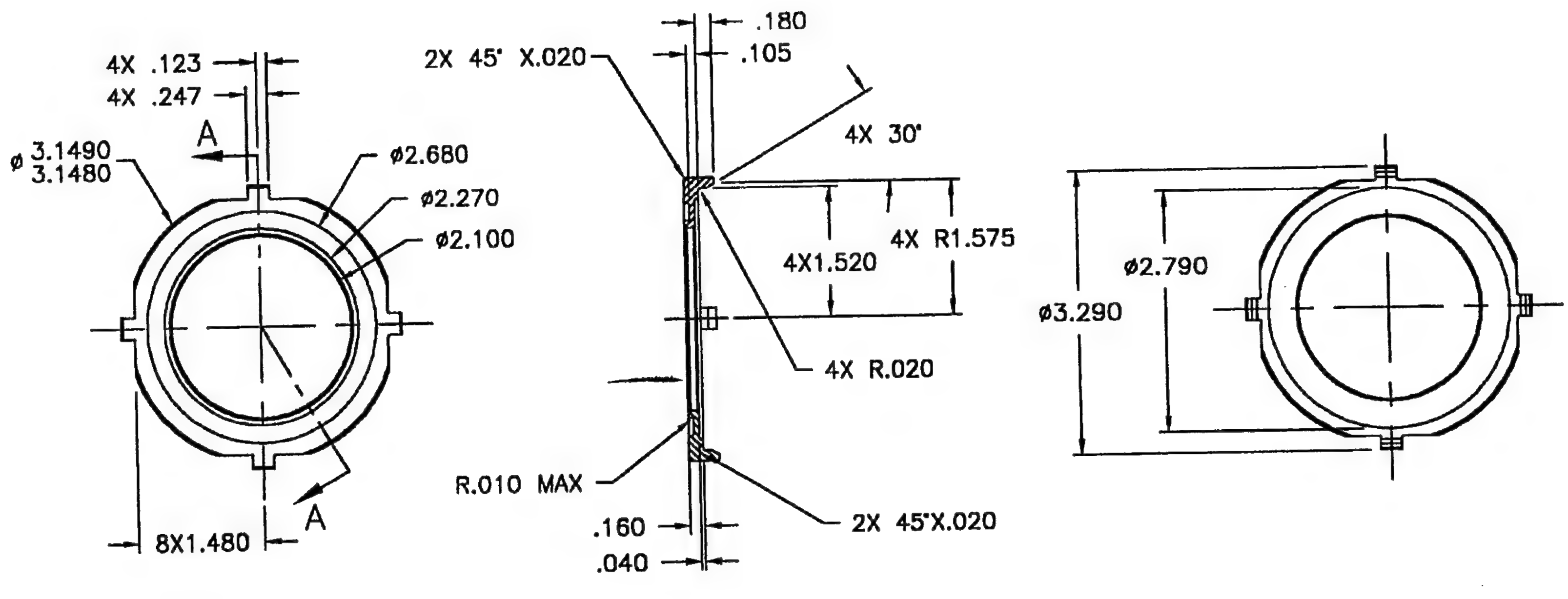

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AUG 7, 1995
EPILOGICS ENGR. DEPT.

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B1	FORMATTED	8-12-95 DON TRINH
A1	ORIGINAL RELEASE	11-25-94 F.FITZ

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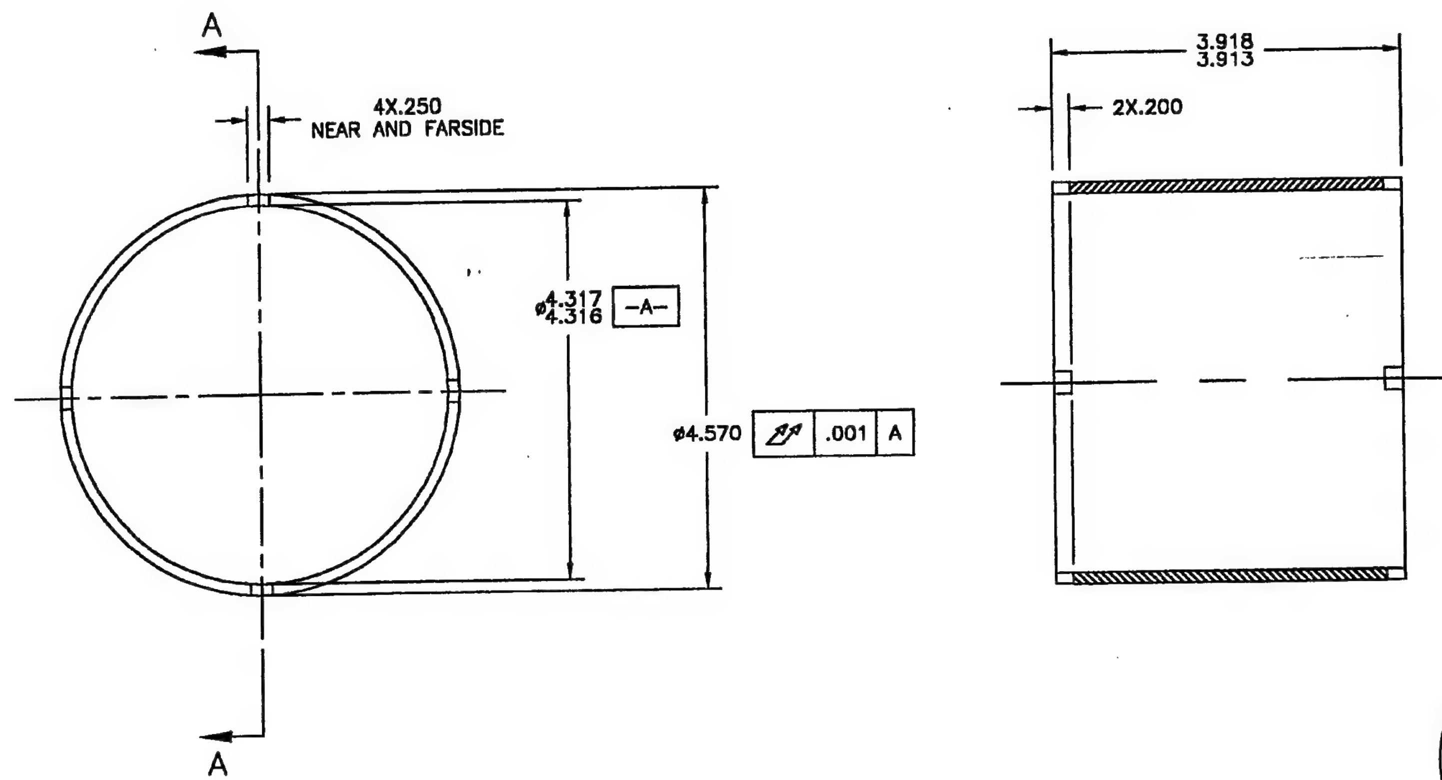

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EPILOGICS ENGR. DEPT.

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NOTES: UNLESS OTHERWISE SPECIFIED.

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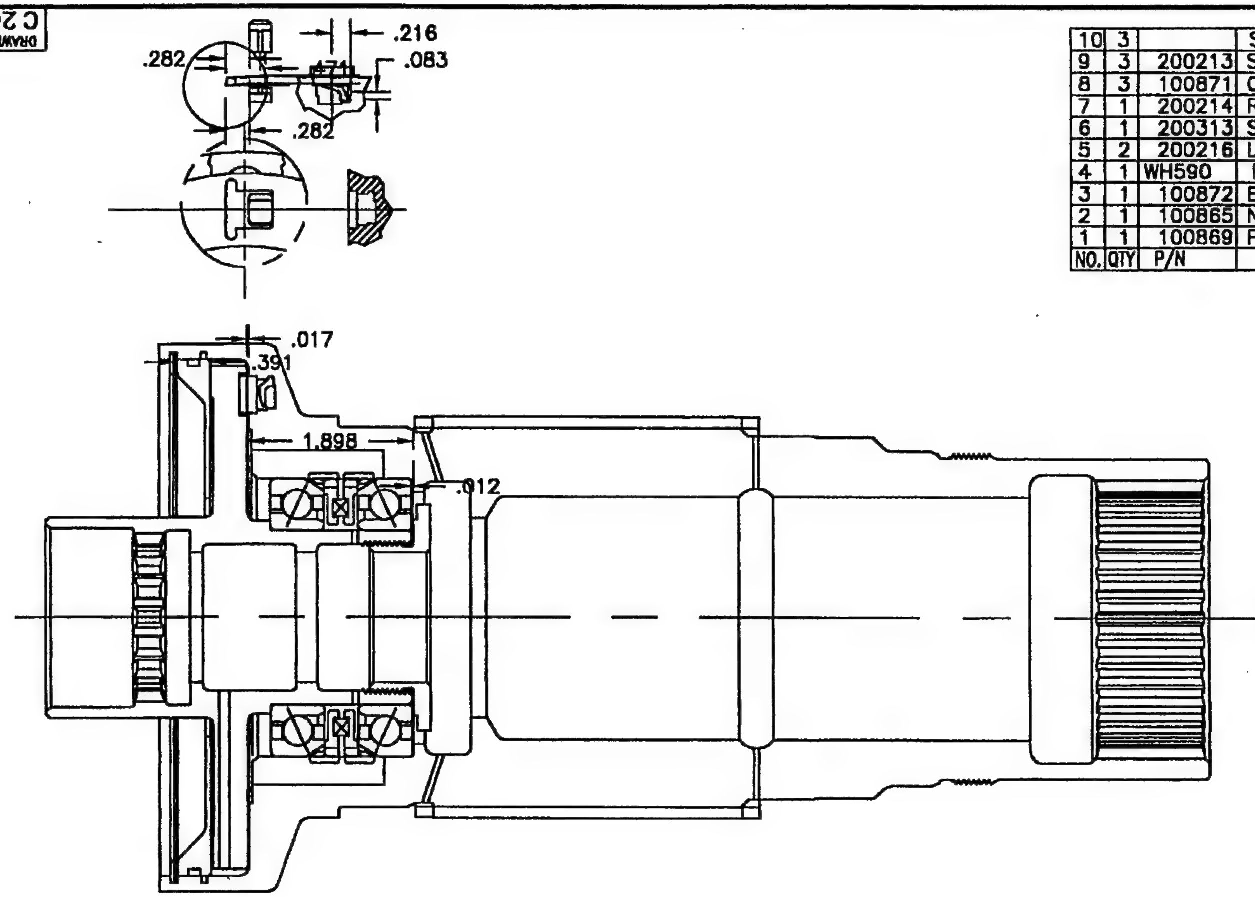
DRAWING NO.
C 200314A1



NO.	QTY	P/N	DESCRIPTION
10	3		SPRING
9	3	200213	STRUT
8	3	100871	CAM
7	1	200214	RETAINER
6	1	200313	SPACER
5	2	200216	LOCK WASHER
4	1	WH590	RET. RING,SMALLEY WH-590
3	1	100872	BACKING RING
2	1	100865	NOTCH PLATE
1	1	100869	POCKET PLATE

DESTROY ALL PREVIOUS DRAWINGS
AUG 10, 1995
EPILOGICS ENGR. DEPT.

NOTES: UNLESS OTHERWISE SPECIFIED.

1. INTERPRET DRAWING PER ANSI Y14.5M.
2. COMPUTER GENERATED DRAWING. SOFTWARE ON FILE IN DOCUMENT CENTER UNDER THIS PART NUMBER.

SYM	REVISIONS	DATE
A1	ORIGINAL RELEASE	8-10-95 DON TRAN

MATERIAL

HEAT TREAT

TOOL CODE

ANGULAR DIMENSION ±1°
FRACTIONS ±1/32
TWO PLACE DECIMALS ±.010
THREE PLACE DECIMALS ±.005
UNLESS OTHERWISE NOTED

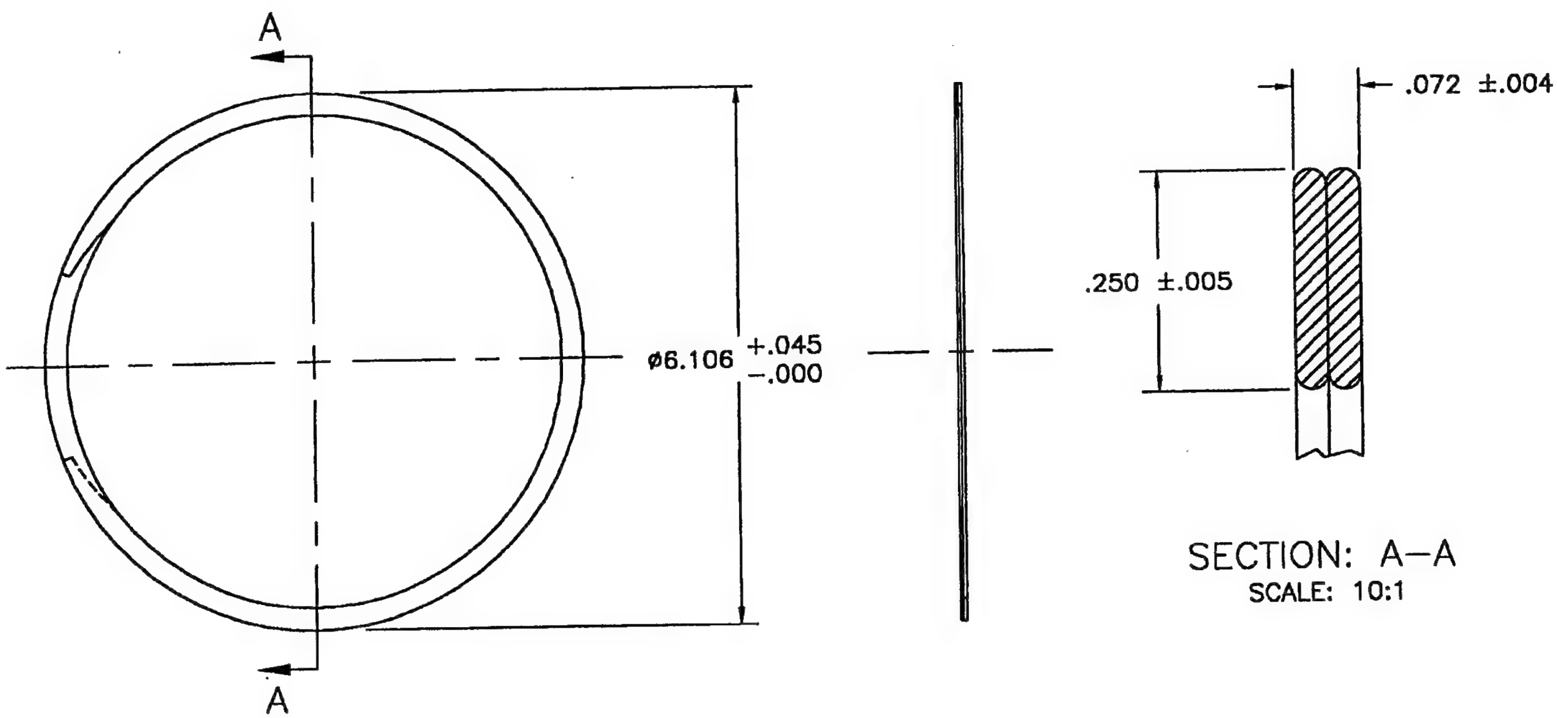
DATE 8-10-95
DRAWN BY DON TRAN
CHECKED BY
SCALE 1:1
PART NO. 200314

EPILOGICS

ASSEMBLY
HIGH SPEED
MD TYPE 1

DRAWING NO. (CAD FILE 200314A1)
C 200314A1

DRAWING NO. C200395A1



NOTES: UNLESS OTHERWISE SPECIFIED.

1. INTERPRET DRAWING PER ANSI Y14.5M.
2. COMPUTER GENERATED DRAWING. SOFTWARE ON FILE IN DOCUMENT CENTER UNDER THIS PART NUMBER.

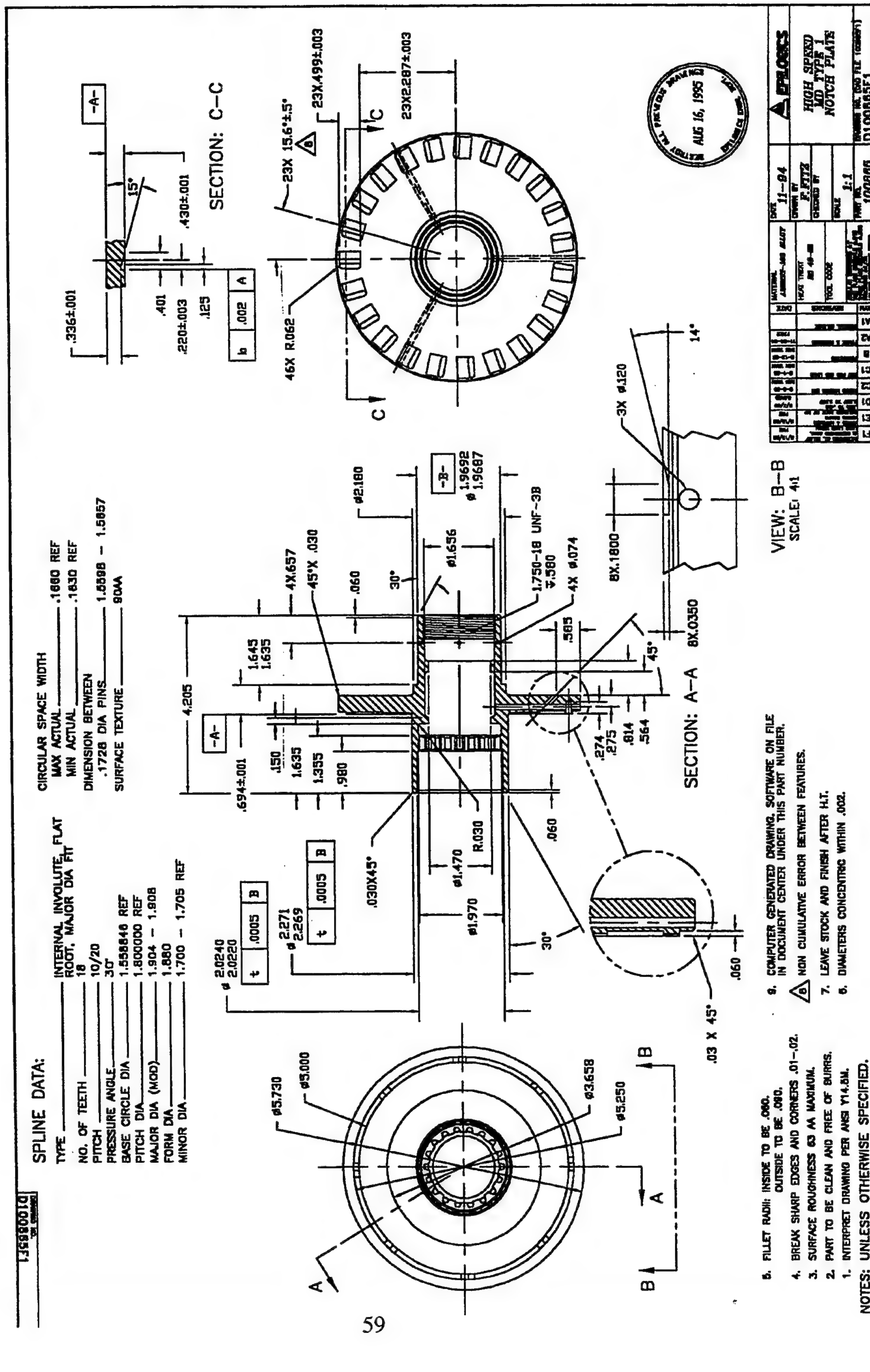
DESTROY ALL PREVIOUS DRAWINGS
JUNE 12, 95
EPILOGICS ENGR. DEPT.

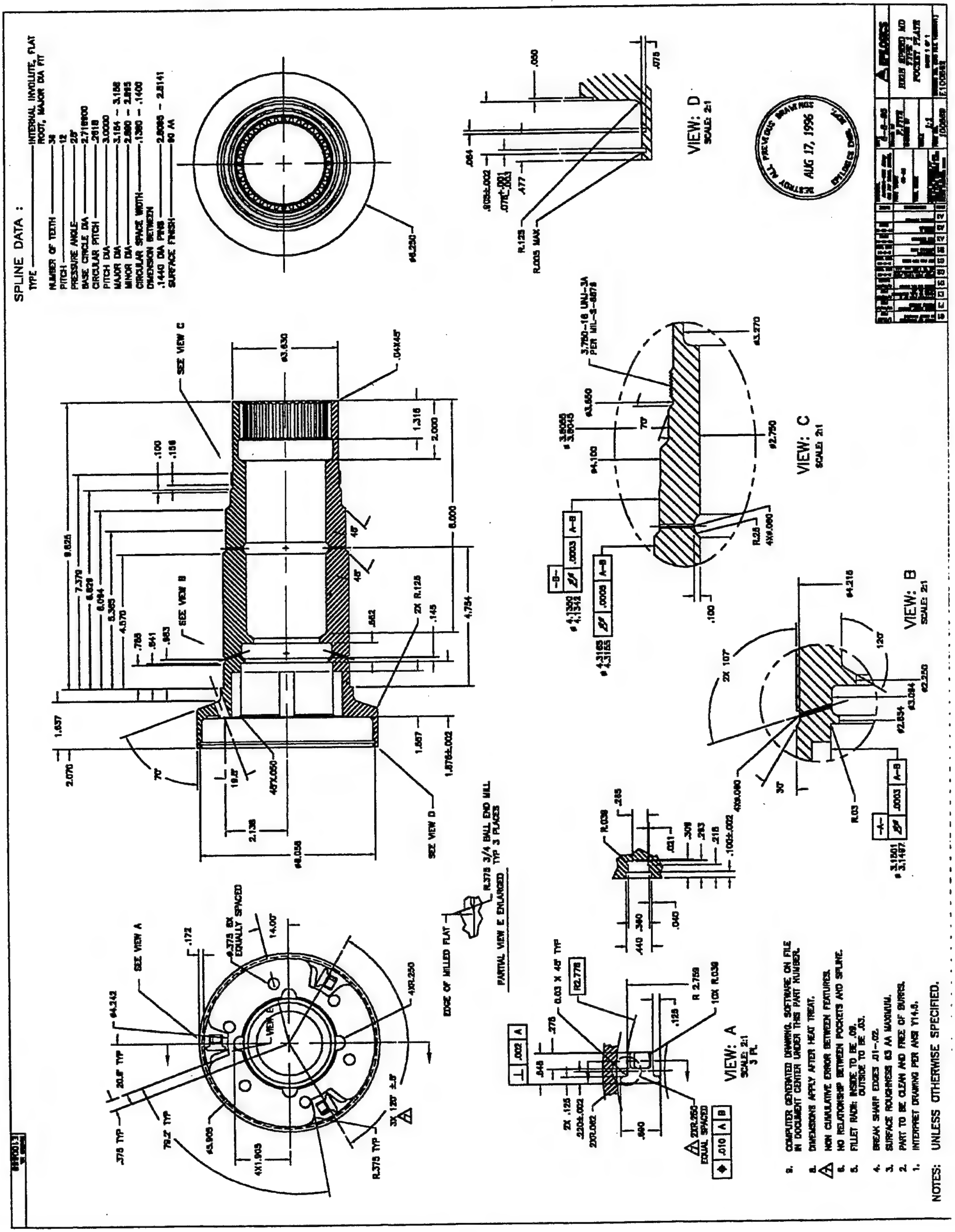
MATERIAL: STEEL	DATE: 11–94	EPILOGICS	
HEAT TREAT	DRAWN BY: F. FITZ	HIGH SPEED MD TYPE 1 SMALLEY RETAINING RING	
TOOL CODE	CHECKED BY		
ANGULAR DIMENSION ±1° FRACTIONS ±1/32 TWO PLACE DECIMALS ±.010 THREE PLACE DECIMALS ±.005 UNLESS OTHERWISE NOTED	SCALE: 1:1		
	PART NO. 200395	DRAWING NO. (CAD FILE 200395A) C 200395A1	

SYM	REVISIONS	DATE
A1	ORIGINAL RELEASE	11-21-94 FITZ
B1	REVISED TO 200395B1	6-12-95 DON TRAN

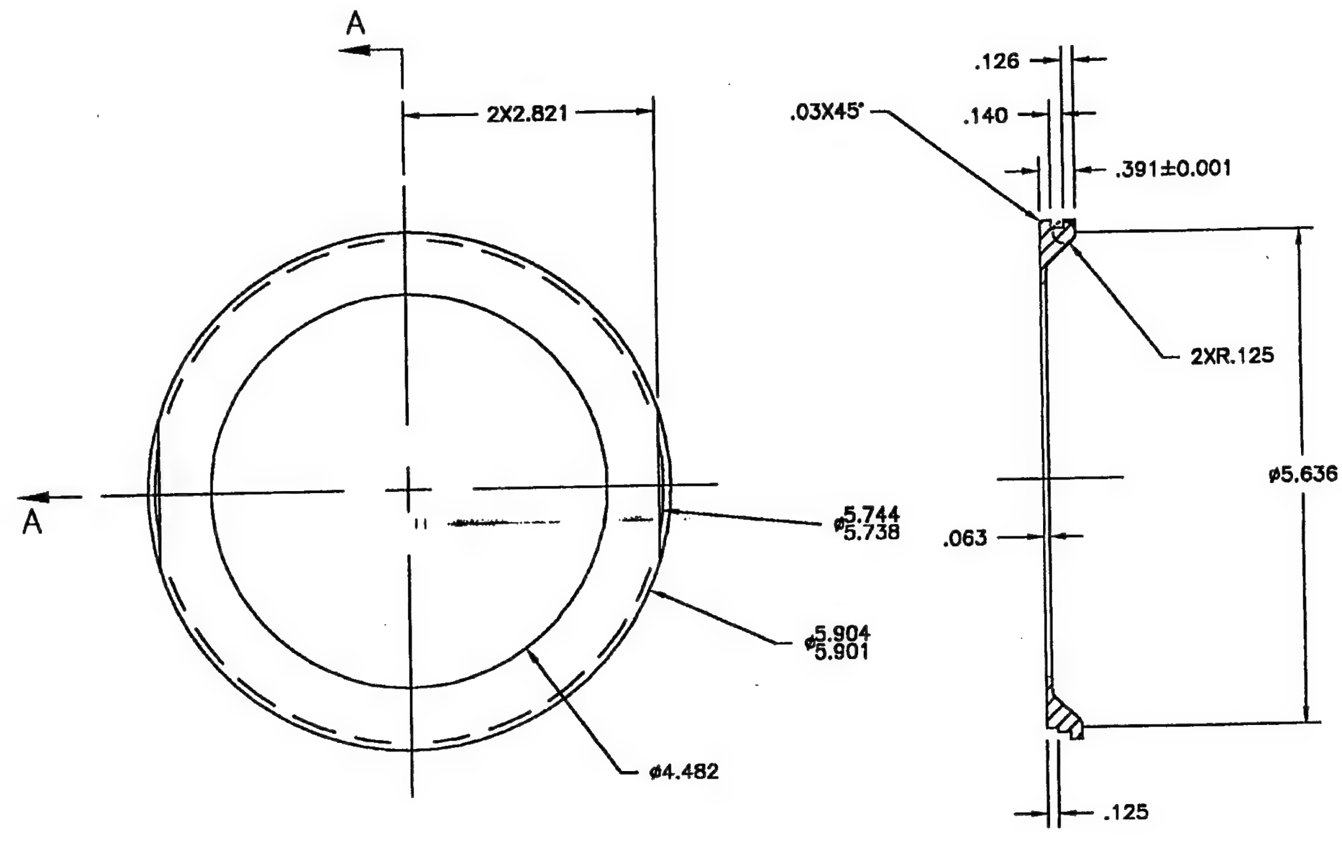
Appendix C

Final Version
High Speed Overrunning Prototype
Drawings


SPLINE DATA:
TYPE — INTERNAL INVOLUTE, FLAT ROOT, MAJOR DIA FIT
NO. OF TEETH — 18
PITCH — 10/20
PRESSURE ANGLE — 30°
BASE CIRCLE DIA — 1.558846 REF
PITCH DIA — 1.800000 REF
MAJOR DIA (MOD) — 1.904 - 1.908
FORM DIA — 1.880
MINOR DIA — 1.700 - 1.705 REF
CIRCULAR SPACE WIDTH
MAX ACTUAL — .1660 REF
MIN ACTUAL — .1630 REF
DIMENSION BETWEEN .1728 DIA PINS — 1.5888 - 1.5857
SURFACE TEXTURE — 90A
SECTION: A-A
SECTION: C-C
VIEW: B-B
SCALE: 4:1
46X R.062
23X 15.6°±.5°
23X.499±.003
23X2.287±.003
AUG 16, 1995
HIGH SPEED MD TYPE 1 NOTCH PLATE
D100885F1
NOTES: UNLESS OTHERWISE SPECIFIED.
1. INTERPRET DRAWING PER ANSI Y14.5M.
2. PART TO BE CLEAN AND FREE OF BURRS.
3. SURFACE ROUGHNESS 63 AA MAXIMUM.
4. BREAK SHARP EDGES AND CORNERS .01-.02.
5. FILLET RADII: INSIDE TO BE .060. OUTSIDE TO BE .060.
6. DIAMETERS CONCENTRIC WITHIN .002.
7. LEAVE STOCK AND FINISH AFTER H.T.
8. NON CUMULATIVE ERROR BETWEEN FEATURES.
9. COMPUTER GENERATED DRAWING, SOFTWARE ON FILE IN DOCUMENT CENTER UNDER THIS PART NUMBER.


SPLINE DATA :
TYPE — INTERNAL INVOLUTE, FLAT ROOT, MAJOR DIA FIT
NUMBER OF TEETH — 36
PITCH — 12
PRESSURE ANGLE — 25°
BASE CIRCLE DIA — 2.718900
CIRCULAR PITCH — .2618
PITCH DIA — 3.0000
MAJOR DIA — 3.154 - 3.156
MINOR DIA — 2.880 - 2.895
CIRCULAR SPACE WIDTH — .1380 - .1400
DIMENSION BETWEEN .1440 DIA PINS — 2.8086 - 2.8141
SURFACE FINISH — 90 AA
VIEW: A
SCALE: 2:1
3 PL
VIEW: B
SCALE: 2:1
VIEW: C
SCALE: 2:1
VIEW: D
SCALE: 2:1
SEE VIEW A
SEE VIEW B
SEE VIEW C
SEE VIEW D
PARTIAL VIEW E ENLARGED
EDGE OF MILLED FLAT
R.375 3/4 BALL END MILL TYP 3 PLACES
3.750-16 UNJ-3A PER MIL-S-8879
DESTROY ALL PREVIOUS DRAWINGS
AUG 17, 1996
NOTES: UNLESS OTHERWISE SPECIFIED.
1. INTERPRET DRAWING PER ANSI Y14.5.
2. PART TO BE CLEAN AND FREE OF BURRS.
3. SURFACE ROUGHNESS 63 AA MAXIMUM.
4. BREAK SHARP EDGES .01-.02.
5. FILLET RADII: INSIDE TO BE .09, OUTSIDE TO BE .03.
6. NO RELATIONSHIP BETWEEN POCKETS AND SPLINE.
7. NON CUMULATIVE ERROR BETWEEN FEATURES.
8. DIMENSIONS APPLY AFTER HEAT TREAT.
9. COMPUTER GENERATED DRAWING, SOFTWARE ON FILE IN DOCUMENT CENTER UNDER THIS PART NUMBER.

DRAWING NO.
C 100872F1



SECTION: A–A

DESTROY ALL PREVIOUS DRAWINGS
JUN 04, 1996
EPILOGICS ENGR. DEPT.

NOTES: UNLESS OTHERWISE SPECIFIED.

1. INTERPRET DRAWING PER ANSI Y14.5M.
2. BREAK SHARP EDGES AND CORNRES .015 MAX.
3. SURFACE ROUGHNESS 63 AA MAXIMUM.
4. COMPUTER GENERATED DRAWING. SOFTWARE ON FILE IN DOCUMENT CENTER UNDER THIS PART NUMBER.

SYM	REVISIONS	DATE
A1	ORIGINAL RELEASE	
A2		
B		
C		
D		
D2		
E1		
F1		
G1		

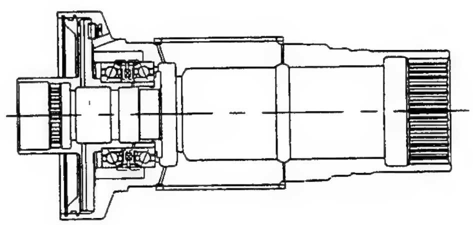
MATERIAL S-7 TOOL STEEL	DATE 11–1994	EPILOGICS
HEAT TREAT RC 48–52	DRAWN BY F. FITZ	HIGH SPEED MD TYPE 1 BACKING RING
TOOL CODE	CHECKED BY	
ANGULAR DIMENSION ±1° FRACTIONS ±1/32 TWO PLACE DECIMALS ±.010 THREE PLACE DECIMALS ±.005 UNLESS OTHERWISE NOTED	SCALE 1:1	
	PART NO. 100872	DRAWING NO. (CAD FILE 100872F1) C100872F1

Appendix D

Charts from the Government/Industry Briefing

High Speed, Lightweight Overrunning Clutch For Rotorcraft

Contract #NAS3-27387



Epilogics, Inc

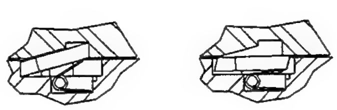
The Epilogics Mechanical Diode

- A planar, high resolution ratchet type OWC
- Positive locking, high torsional stiffness
- Oil managed, "non-contact" overrunning
- Proven in racing and passenger car applications

Epilogics, Inc

MD Operation

- Simple geometry
- Low stresses



LOCKED OVERRUNNING

Epilogics, Inc

Phase 1

- Developed two primary design concepts
- Selected one design for further development
- Completed design calculations

Epilogics, Inc

Phase 2

Prototype and Evaluate at BHTI

- Constructed initial prototypes based on Phase 1 design
- Performed nominal overrun testing of the prototypes
- Revised the design based on test results and tester geometry
- Fabricated test prototypes
- Performed overrun testing and design modifications
- Performed load cycle testing

Epilogics, Inc

Project Discussion

- The Design
  - Intended Operation
  - Critical Aspects and Calculations
- Development
  - Initial Overrun Testing
  - Subsequent Modifications and Results
  - Load Cycle Testing

Epilogics, Inc

Design Overview

- Intended Operation of the "Helicopter" MD
- A More Detailed Look at the Design

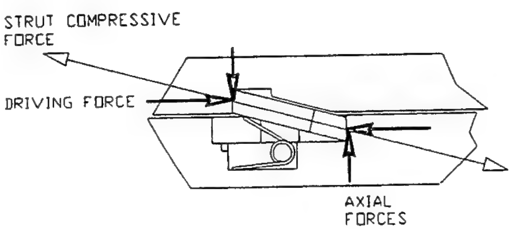
Epilogics, Inc

Mechanical Diode Essentials

- How the Strut Transfers Torque
- How we Ensure that the Strut is Biased and will Cam into the Notch
- How the Counterweight Provides High Speed Strut Biasing
- Lubrication

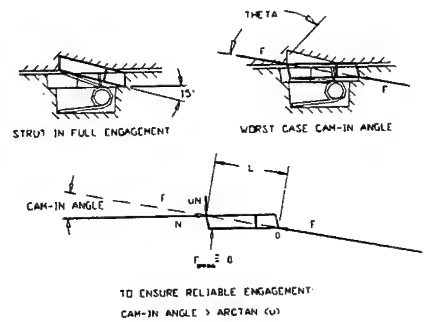
Epilogics, Inc

How the Strut Transfers Torque



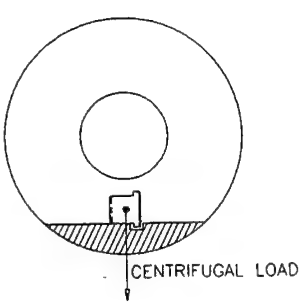
Epilogics, Inc

How we Ensure the Strut Cams into the Notch



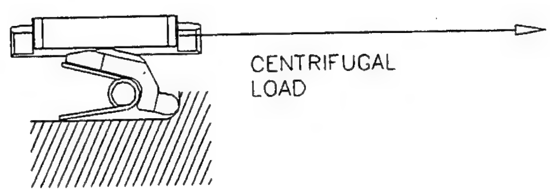
Epilogics, Inc

The Need for a Counterweight



Epilogics, Inc

How the Counterweight Provides Strut Biasing



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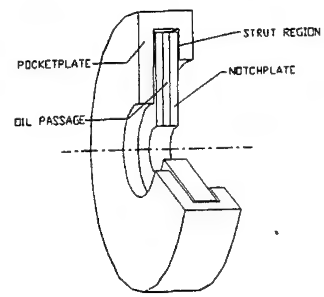
MD Lubrication

- The strut needs to stay submerged
- Flow is required to prevent overheating of lubricant

Epilogics, Inc

Lubrication: How it's done



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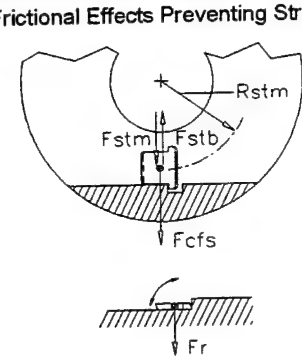
Some Further Detail

- Frictional Effects Preventing Strut Rotation
- Counterweight Design
  - Force Resolution
  - Stress
- Ultimate Strength
- Fatigue Life
- Fluid Drag
- Horsepower Losses

Epilogics, Inc

Friction

Frictional Effects Preventing Strut Rotation



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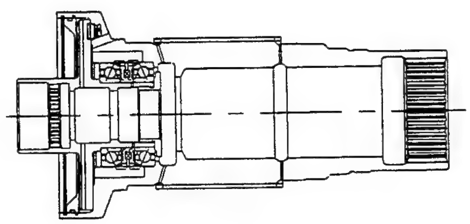
Frictional Effects on Strut

Frictional Effects Preventing Strut Rotation



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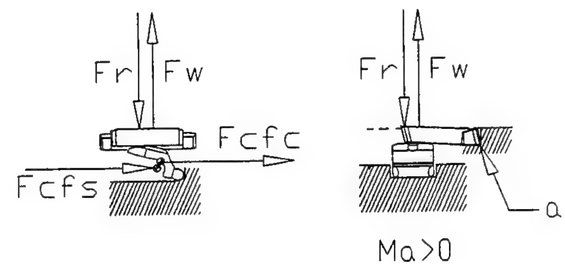
Forces Affecting the Strut and Counterweight

Counterweight Design



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Counterweight Contact Stress

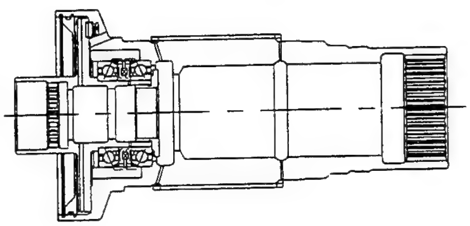
Counterweight Design

- Contact Stress at Pivot
  - Max stress=.591*((Fn/L*E)/Kd)^.5
    - Kd=(D1*D2)/(D1+D2)
- 27,454 psi

Epilogics, Inc

Initial Tests

We discover that the MD overheats during sustained overrunning at moderate speeds



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Initial Test Observations

- Oil temperature rise through the MD was too high
- There was some evidence of excessive heat on the notch plate back
- Struts and notches showed excessive and uneven wear

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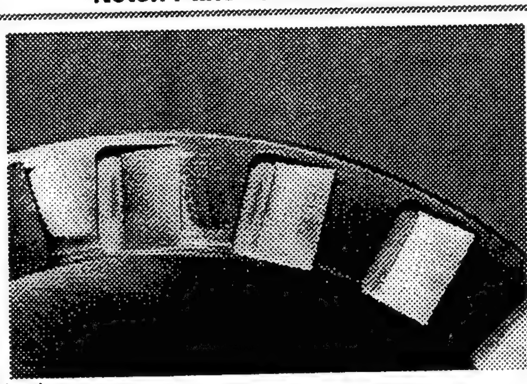
Notch Plate - Initial Version



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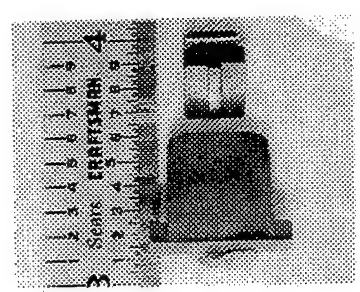
Strut & Counterweight - Initial Design



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First Modifications

- Reduced the oil shear area on the back of the notch plate
- Recontoured the counterweights for contact closer to the strut centers
- Reduced the overhung mass of the counterweights
- Increased the ID of the back up plate to reduce trapped oil volume

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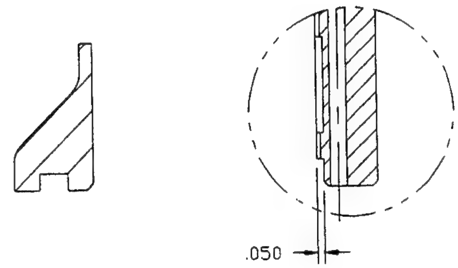
First Modification Details



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Second Test Series

Decreasing oil shear reduces the oil exit temperature

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Second Test Series Observations

- Decreasing oil shear decreased oil temperature to 60% of initial values
- Recontouring counterweights made strut wear more uniform
- Oil temperature rise is still too large for operation at 20,000 rpm
- Struts and counterweights continue to wear during overrunning

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Second Set of Design Modifications

- Increase shear relief on back of notch plate
- Decrease the width of the notch plate locating ring
- Add oil wedge forming ramps to the notch plate locating ring oil grooves
- Relieve surface of pocket plate except for pocket "pedestals"
- Drill oil drain holes in pocket plate to limit the oil shearing ring
- Make counterweights out of Titanium alloy to reduce strut force

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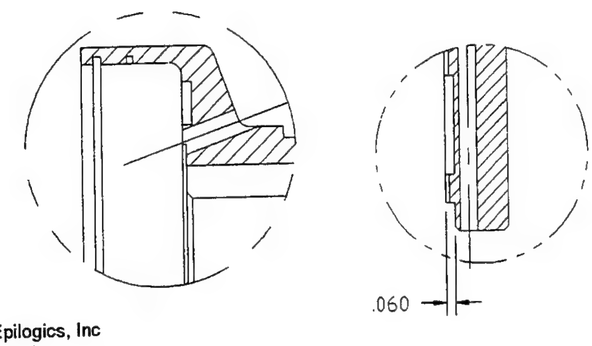
Second Modification Details



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Third Test Series

Further reductions in oil shear improve oil temperature

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Third Test Series Observations

- Reduction of oil shear opportunities further reduced oil temperature rise
- Titanium counterweights further reduced strut wear during overrunning
- Although improved, oil temperatures were still too high for over 20,000 rpm overrunning operation

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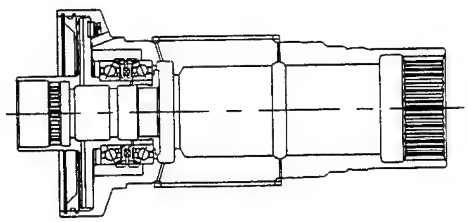
MD Ultimate Strength

Ultimate Strength



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Tangent-Modulus Method

Ultimate Strength

σcr=Pcr/A=(π^2*E*I)/L^2*A, Euler

provided that σcr<σpl

σcr=Pcr/A=(π^2*Et*I)/L^2*A, Engesser

provided that σcr<σy

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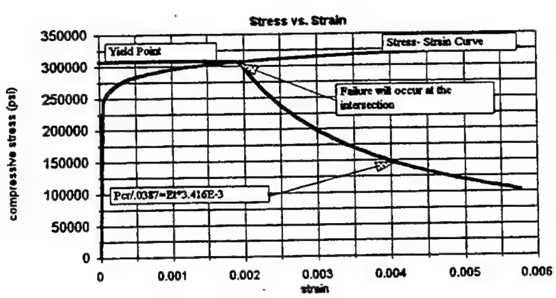
Results

Ultimate Strength



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Max Torque=2473 ft-lbf

MD Fatigue Life

Fatigue Life

- Typical Fatigue Crack Initiation Sites
  - High Cycle Fatigue
    - Ring groove
    - Pocket Relief Corners and Edges
  - Low Cycle Fatigue
    - Ring Groove
    - Notch Relief Corners and Edges

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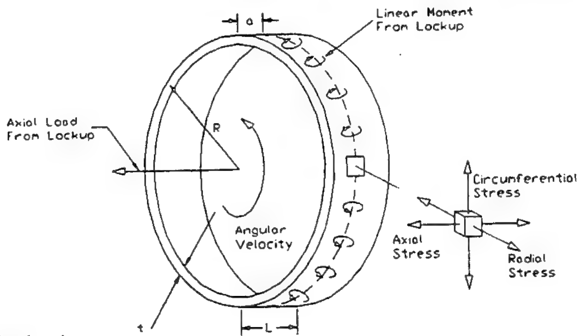
Ring Groove Fatigue

Fatigue Life



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Principle Stresses

Fatigue

- Circumferential Stress (0-70,000 psi)
  - Dynamic Stress from Rotation
- Radial Stress (0-100 psi)
  - Dynamic Stress from Rotation
- Axial Stress (0-3,000psi)
  - Axial Stress from Lockup Load
  - Bending Stress

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Fluid Drag on a Spinning Disk

Fluid Drag

- Fluid Drag Induced Moment
  - M=Cm(.5τ*ω^2*Rn^5)
    - where:
    - Rn=Reynolds #
    - Cm=Moment Coefficient
    - ω=Angular Velocity

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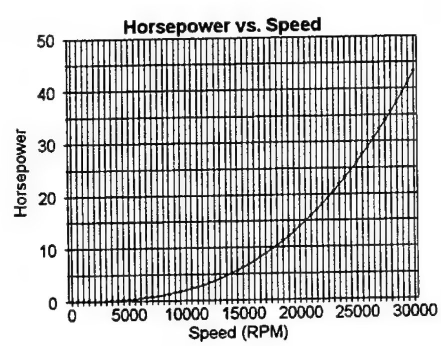
Resulting Horsepower Losses



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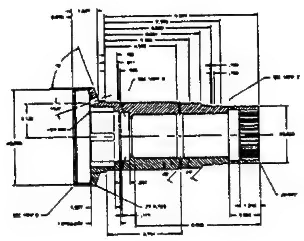
High Speed Overrunning Development



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Overrun Test Plan Objectives

Developed by BHTI

- Overrunning at maximum temperature & highest speed differential
- Engagement at speed after overrunning
- Engagements at cold temperatures
- Fail-safe failure (no lock-up) during 30 minute loss of lube test

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Test Parameters Monitored

- Time of Day
- Total Test Time
- Test Cell Ambient Temperature
- Input Shaft Speed
- Output Shaft Speed
- Oil Inlet Temperature
- Oil Outlet Temperature (2 places)
- Oil Inlet Flow Rate and Pressure
- Support Bearing Temperatures

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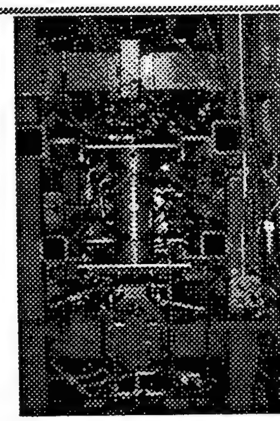
BHTI Test Stand



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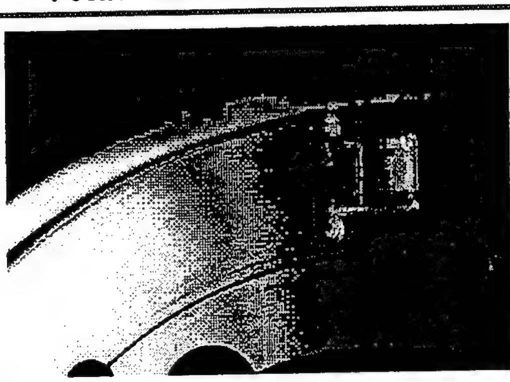
Pocket Plate - Second Mod Series



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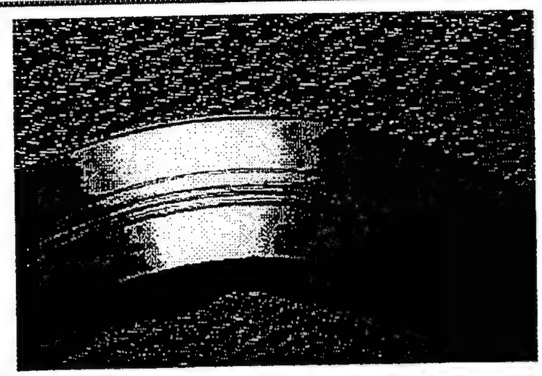
Backing Plate - First & Second Mod Series



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Third Set of Design Modifications

- Increase the size of the oil drain holes in the pocket plate
- Use AerMet-100 notch plates with all previous modifications
- Surface harden struts and pocket plate by nitriding
- Build a special notch plate with angled oil feed holes

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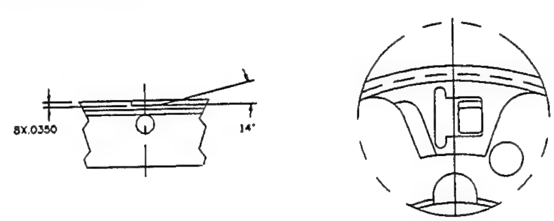
Third Modification Details



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Fourth Test Series

Good Performance to Limits of Test Stand

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Fourth Test Series Observations

- Thermal behavior was improved, probably acceptable
- Wear degree and uniformity was improved, probably manageable with material changes except for:
- Spring wear remained surprisingly high; needs improvement

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Pocket Plate - Third Mod Series

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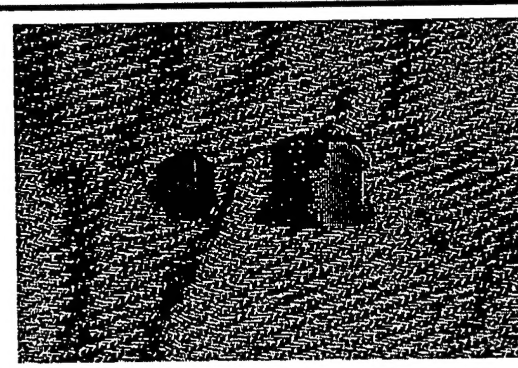
Counterweight & Strut - Third Mod Series



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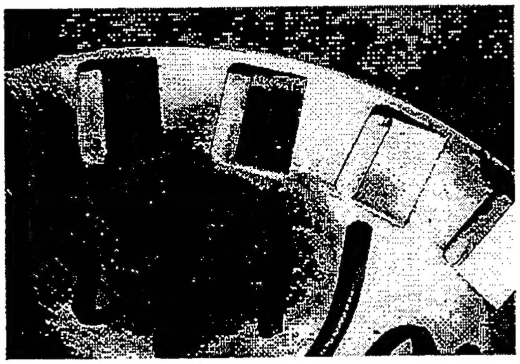
Notch Plate - Third Mod Series



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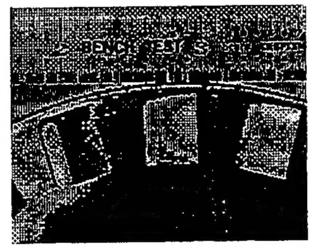
Notch Plate - Third Mod Series



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Fatigue Life Testing

Test Plan

- Load cycle at maximum continuous torque
  - 875 ft-lb applied moment each cycle
  - Cycle at 2 cycles per second
  - Duration - 20,000 cycles or failure
- Load cycle at the limit torque
  - 1750 ft-lb applied moment each cycle
  - Cycle at 3 or more cycles per second
  - Duration - 1,000,000 cycles or failure

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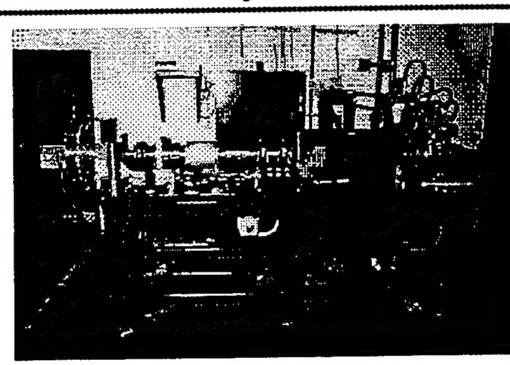
Load Cycle Tester



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Fatigue Test Results

- Load cycle at maximum continuous torque
  - 20,000 cycles completed
  - No evidence of distress on struts, pockets or notches
  - Minor fretting corrosion on the MD bearing housing
- Load cycle at the limit torque
  - Fixture failed at 2,091 cycles
  - Made new fixture with integral, heat treated spline
  - Completed 296,143 cycles, notch plate shaft failed
  - No evidence of distress on struts, pockets or notches

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Strut - 20,000 Load Cycles

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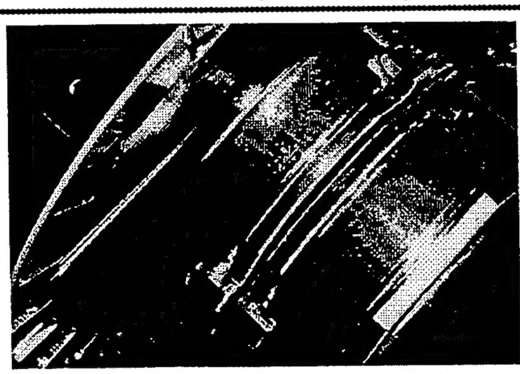
Notch Plate - 20,000 Load Cycles



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Conclusions

- MD type one-way clutches can overrun and engage reliable at speeds up to 20,000 rpm
- Additional development is required to reduce counterweight and spring wear to acceptable levels

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Frank Fitz and Craig Gadd

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Epilogics, Inc.
160C Albright Way
Los Gatos, California 95030

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13. ABSTRACT ***(Maximum 200 words)***

The objective of this program was to evaluate the feasibility of a very high overrunning speed one-way clutch for rotorcraft applications. The high speed capability would allow placing the one-way clutch function at the turbine output shaft, that is, the input of the rotorcraft's transmission. The low drive torque present at this location would allow design of a relatively light one-way clutch. During the course of this program, two Mechanical Diode (MD) type overrunning clutches for high speeds were designed. One of the designs was implemented as a set of prototype clutches for high speed overrun testing. A high speed test stand was designed, assembled and qualified for performing overrunning and engagement tests at speeds up to 20,000 rpm. MD overrunning clutches were tested at moderate speed, up to 10,000 rpm and substantial thermal problems associated with oil shear were encountered. The MD design was modified, the modified parts were tested, and by program end, clutches were tested in excess of 20,000 rpm without excessive lubricant temperatures. Some correctable wear was observed and remains as a clutch characteristic which needs further improvement. A load cycle tester with a special, long, sample section was designed, built and then prototype clutches were fatigue tested to verify that the clutch design was suitable for carrying the specified power levels.

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